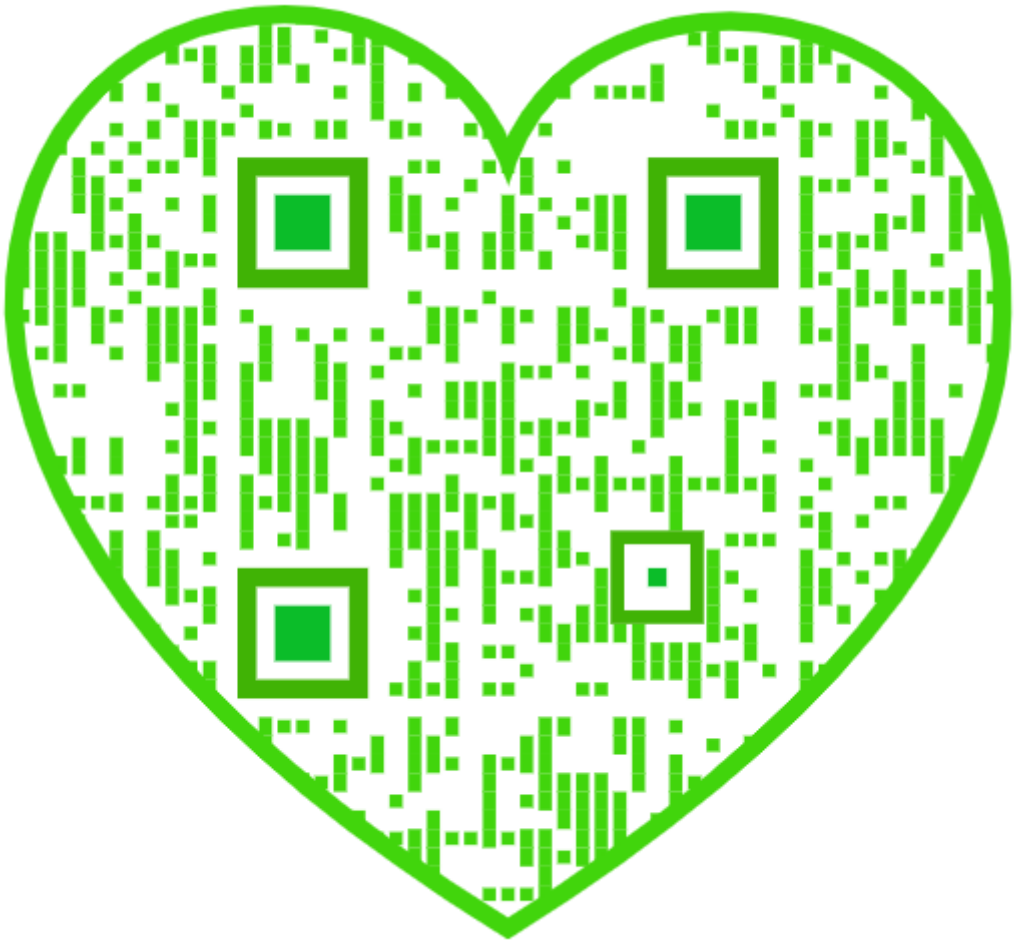

# HANDBOOK
OF
# MECHANICAL REFRIGERATION

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OF

# MECHANICAL REFRIGERATION

BY

H. J. MACINTIRE

*Professor of Refrigeration in the University of Illinois*

NEW YORK

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# PREFACE

In the author's opinion an engineering handbook must naturally include all information an engineer is likely to be in need of in the particular field under discussion. It should have some theory as the basis for the fundamental formulæ, and it certainly must have all information required for the comprehension of the design and the practical operation of any system falling within the field of the handbook. In the present case, therefore, there is given a brief treatment of certain thermodynamic material in the expectation that for more detailed study of this phase of the subject the reader will consult a text on thermodynamics; also, such information concerning electric motors, steam and oil engines, and related matters as is absolutely essential for the grasp of the main material.

Much of the material of this book has been gathered and used for teaching purposes. This is particularly true of the special applications of refrigeration, a division of the subject which, therefore, has been treated perhaps more fully in this book than has heretofore been attempted. The material on compressors and on the absorption machine has been brought up to date, and it is thought that the chapters on cold storage and ice manufacture will be found particularly complete and of practical value. A recent great increase in the use of the household refrigerating machine and the so-called ice cream cabinet has materially assisted in the development of the automatic machine, while the more or less general fear of ammonia has been the incentive towards the use of some safer refrigerant such as methyl chloride and the hydrocarbons. We have more and more such things as "manufactured weather," the use of precooling and refrigeration during transit, district cooling, hotel and apartment house refrigeration. All of these subjects will be found treated in the chapter on cold storage. The development of the raw water ice manufacture has stimulated the study of water treatment and air agitation while the increasing cost of power and labor has necessitated the use of the Diesel oil engine, the electric motor and labor-saving devices in the tank room. Material, therefore, on these matters also will be found in this volume.



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Wherever possible, illustrative examples have been worked out. As a rule such illustrative problems are very similar one to another, and vary only in the physical constants used. In some special cases these constants are not known accurately (as, for example, the heat capacities of distillates which vary in the different oil fields). Not many years ago Professor Denton was forced to find out experimentally for himself certain properties of ammonia before he could work up his tests; and the early refrigerating engineers experimented on the specific heat of various commodities, such as plowshares, brines, etc., before calculating the refrigeration required in special problems. In some cases similar procedure is still necessary, especially in certain chemical processes, but through the efforts of the American Society of Refrigerating Engineers, the United States Bureau of Standards, and other societies, refrigerating data are being rapidly collected and distributed. Where such information is already available the author has endeavored to embody it in this treatise.

An attempt has been made to give the authority for all material not the author's own, and to include references whenever possible. In particular it should be stated that the introduction follows closely certain parts of the "Principles of Thermodynamics," by Professor G. A Goodenough, to whom thanks are also due for many helpful suggestions. The manufacturers of compressors and refrigerating equipment have been very generous in supplying information without which a book of this character would not be possible. To these engineers, and in particular to his former students, now engaged in the refrigerating industry, the author desires to express his thanks. He wishes also to thank Mr. R. E. Gould, research assistant in the Engineering Experiment Station at the University of Illinois, for his reading of the page proof.

With a book of this nature, especially in the first edition, it is practically impossible to eliminate errors. The author solicits information regarding any defects in the problems or the text.

H. J. Macintire

Urbana, Illinois

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# HANDBOOK OF REFRIGERATION

## CHAPTER I

### INTRODUCTION

**Mechanical Refrigeration** is one of the many applications of the theory of heat energy, but it is a reversed cycle as compared with that used in prime movers. In mechanical refrigeration heat is absorbed at the lower temperature and is rejected at the upper temperature, and a pump—the compressor or other device of similar nature—is employed to make the cycle possible. A book on refrigeration has to cover primarily the field of its application, but it needs also to give material allied with the industry and which the refrigerating engineer finds need of in his work as well as the fundamental basis for the cycles and the formulæ used. In this chapter, therefore, these related matters will be taken up very briefly, and an explanation of the diagrams used in representing the refrigerating cycle will be given.

**Energy.**—A moving body, a compressed spring, or a substance like steam is said to possess energy when it has the ability to perform work while its condition undergoes a change. This energy may be mechanical *kinetic* energy due to its motion, or mechanical *potential* energy due to its position or its state of aggregation. When the energy of a substance is due to the motion or the change or arrangement of the molecules this energy is called *intrinsic* energy.

**Temperature.**—Temperature is an indication of the rapidity of the vibration of the molecules of a body, and usually a change in the intensity of this vibration is evidenced by a change in the volume of the body, as when heated air expands or cooled mercury contracts. There are two scales of temperature in common use, one in which the distance between the melting and boiling points of water is designated as a hundred divisions (with the zero and the one hundred at these two points, respectively), and the other naming 32 as the freezing and 212 as the boiling temperature with 180 divisions between. In the United States the

latter, the Fahrenheit scale, is used with practical unanimity by mechanical engineers; while the other, the Centigrade scale, is used in Great Britain and on the Continent. The *absolute zero* of temperature, as is evident from what precedes, is the condition where there is no vibration of the molecules of the body. This point is considered to be 459.7 degrees below the zero on the Fahrenheit scale.

**Pressures.**—Pressures may be measured in inches of water, or of mercury, in atmospheres, or in kilograms per square centimeter. The custom in the United States is to use as a unit the pound per square inch. The ordinary gage records in pressures above the atmosphere, so that the *absolute pressure* is obtained by adding to the gage reading the equivalent of the atmospheric pressure.

**Heat.**—It has been found by experiment that if two bodies of different temperature are placed near one another heat will flow from the hotter to the colder body until finally the same temperature is attained by each. The heat in a body may be *sensible* heat (indicated by its hotness or lack of hotness) or it may be *latent* heat, a term used to denote the change of aggregation of the molecules such as is evident when the substance changes from a solid to a liquid, from a liquid to a gas, or vice versa. The unit of heat is taken as the British thermal unit (B.t.u.) which is defined as $\frac{1}{180}$ of the amount of heat required to raise one pound of water from 32 to 212 deg. F.

There are other forms of energy besides heat energy: such as, chemical and electrical energy. By numerous experiments it has been demonstrated that energy of one kind can be transformed into energy of any other kind.

**The Energy Equation.**—Experiment has proved that if heat energy is supplied to a substance a change may occur in the intrinsic energy, or external work (work performed on an external body) may be done, or both. This is stated as follows:

$$_1Q_2 = U_2 - U_1 + AW$$

where $_1Q_2$ is the heat added during the process *12*, $U_1$ is the initial and $U_2$ is the final intrinsic energy of the substance and $W$ is the work performed in foot-pounds (ft.-lb.). The expression $A$ is for the reciprocal of the mechanical equivalent of heat ($\frac{1}{778}$).

The intrinsic energy may consist of both thermal *kinetic* energy, indicated by the temperature, and thermal *potential* energy, indicated by the state of aggregation of the substance. Energy, like pressure, volume, temperature, etc., being a point function, may be represented by a point on a coordinate plane and is independent of the path by which

the stated condition was obtained. It follows from this that in a complete cycle, on arriving at the initial point, the *energy change* is zero.

**The First Law of Thermodynamics.**—The experiments of Rumford and others have proved that mechanical and heat energy have a certain relationship; that is, when work is expended in producing heat, the heat produced is proportional to the work performed. This can be expressed by the relation:

$$Q = AW$$

where $W$ denotes the work performed, usually expressed in mechanical engineering in foot-pounds (ft.-lb.). The quantity $AW$ would then be in B.t.u.

**Conversion Constants.**—The following conversion constants, based on the first law of thermodynamics, will be of value in the solution of problems.

1.0 hp. = 42.44 B.t.u. per minute.

= 2546 B.t.u. per hour.

1.0 KW. hr. = 3414.5 B.t.u.

1.0 B.t.u. = 777.6 ft.-lb.

**Specific Heat.**—The ability of different substances to absorb heat varies, as instanced by water, copper and iron. The ratio of the capacity for the absorption of heat possessed by equal weights of certain substances, as compared with the capacity of *water*, gives the specific heat ($c$), which is seldom more than unity. In the case of gases and vapors it is necessary to state under what conditions the heat was absorbed, as for example whether it was at constant pressure ($c_p$) or under constant volume ($c_v$).

**The *p-v* Diagram.**—Changes in a substance involving changes in pressure and volume may be represented by a line on the plane of two coordinate axes, as for example the so-called $p$-$v$ plane. It can be shown that the projected area under such a "curve" represents the external work performed during the particular process under consideration. In the case of the complete cycle the area of the closed figure on the $p$-$v$ diagram represents the work performed, and the energy change is zero.

**Changes of State.**—Such changes on the $p$-$v$ plane have various names, depending on the manner in which the volume varies with respect to the pressure: for example, the *isothermal* (constant temperature), the *isobar* (constant pressure), the *constant volume*, and the *adiabatic* (the process during which no heat is added to the substance

nor given up by the substance). The ordinary compression of air or of any other refrigerant during the compression stroke is none of these but is one having the equation $pv^n$ = a constant, where $p$ is the pressure exerted by the gas, $v$ is the volume occupied by the gas in cubic feet, and $n$ is the exponent of the volume, this being usually about 1.4 for air and about 1.28 for ammonia and carbon dioxide. The external work in any of these cases is the projected area on the volume axes of the $p$-$v$ plane. It is convenient in such cases to apply the energy equation

$$_1Q_2 = U_2 - U_1 + AW$$

in order to find the heat absorbed or rejected. Special formulæ are available in each case and these will be taken up in their turn.

**Available Energy.**—When heat energy is converted into work, only a part can be so converted, the amount depending on the temperature range in the cycle of operations and on the separate processes. In other words, if heat is supplied to a heat engine, a part, and sometimes only a very small part, is available for, and can be converted into, useful work. The separate processes making up the cycle of operations may be *reversible* or *irreversible*. The theoretical *frictionless adiabatic* and the *isothermal* are examples of the former type, whereas all practical processes are irreversible. By the term "reversibility" is meant that all the events must be taken in the reverse order and that at the end of the process all conditions, both externally and internally, must be restored to their original state.

**The Second Law of Thermodynamics and the Carnot Cycle.**—The second law of thermodynamics can be stated in a number of ways. From experience, it can be said that:

1. An irreversible change causes a loss of availability.
2. A reversible change causes no change in availability.
3. No increase in available energy in a system can take place as a result of a self-imposed change in the body.

The **Carnot cycle** is one composed of two isothermals and two frictionless adiabatics. If $T_1$ is the absolute temperature of the source and $T_2$ is the absolute temperature of the refrigerator, both in absolute degrees, then it may be proved that the efficiency of the Carnot cycle is:

$$E = \frac{T_1 - T_2}{T_1} = \frac{Q_1 - Q_2}{Q_1} = \frac{AW}{Q_1}$$

where $Q_1$ is the heat supplied from the source, $Q_2$ is the heat rejected to the refrigerator, and $AW = Q_1 - Q_2$ is the work performed by the engine.

The refrigerating cycle is a reversed cycle (reversed as compared with the steam engine cycle) and it absorbs heat at the lower temperature and rejects it at the upper one, the reversed motor (a compressor) making this cycle a possibility.

**Carnot's principle** says that such an engine cannot be more efficient than the direct-acting engine working between the same temperature limits. Two examples of Carnot's principle follow:

**Problem.**—A reversed Carnot engine is used for refrigeration. The temperature of the refrigerator is − 10 deg. F. and that of the hot body is 80 deg. F. Find the horse power required to drive the engine if 10,000 B.t.u. per minute is to be taken from the cold body. The efficiency of the cycle is

$$E = \frac{(80 + 460) - (-10 + 460)}{(80 + 460)} = \frac{90}{540} = 0.1667$$

$$0.1667 = \frac{Q_1 - 10{,}000}{Q_1},$$

and therefore

$$Q_1 = 12{,}000 \text{ B.t.u.}$$

from which

$$AW = 2000 \text{ B.t.u. per minute}$$

$$= 2000 \div 42.44 = 47.1 \text{ hp.}$$

As another example, suppose a reversed Carnot engine is used for heating a building. The outside temperature is 0 deg. F., and the inside of the building is to be maintained at 70 deg. F., and 100,000 B.t.u. per hour are required for heating. Find the minimum horse power required. It seems to be somewhat difficult to conceive such a problem, and to realize that heat can be absorbed from the cold body at 0 deg. F., but similar conditions prevail in every refrigerating plant. The efficiency of the cycle is, as before,

$$E = \frac{T_1 - T_2}{T_1} = \frac{70 - 0}{460 + 70} = 0.132$$

But

$$\frac{Q_1 - Q_2}{Q_1} = 0.132$$

and the problem says that $Q_1$ is 100,000 B.t.u.;

therefore,

$$\frac{100{,}000 - Q_2}{100{,}000} = 0.132,$$

from which

$$Q_2 = 86{,}800 \text{ B.t.u.}$$

and

$$AW = 13{,}200 \text{ B.t.u. per hour,}$$

and

$$13{,}200 \div 2546 = 5.19 \text{ hp.}$$

**Entropy.**—In the study of heat processes and cycles it is desirable to have a workable diagram. The diagram on the $p$-$v$ plane is useful only in a limited manner, as it shows the external work performed and not the manner in which the heat flows. The so-called entropy diagram is one devised to show by the projected area on the entropy axis the amount of heat absorbed or rejected during the process or system. Goodenough[1] defines entropy as a quantity proportional to the unavailable energy. A frictionless adiabatic is one process conceived to have *constant entropy*.

Entropy, being a point function, is expressed by:

$$S_2 - S_1 = \int \frac{dQ}{T} + \int \frac{dH}{T}$$

where $S_2 - S_1$ is the change of entropy, $Q$ is the heat absorbed, $H$ is the heat due to friction, $T$ is the absolute temperature of the substance during the process.

During a frictionless isothermal process

$$S_2 - S_1 = Q \div T \quad \text{and} \quad Q = T(S_2 - S_1) \quad \text{Fig. 1}b$$

Whereas if

$$dQ = Mc_v dt$$

then

$$S_2 - S_1 = Mc_v \int \frac{dt}{T} = Mc_v \log_e \frac{T_2}{T_1}$$

where $c_v$ is a constant.

The temperature-entropy diagram is one with absolute temperature ($T$) as ordinates and entropy as abscissa. These diagrams are easily drawn for vapors as the entropy in such cases is usually given in the tables of the properties of saturated vapors.

**Thermal Potential.**—A quantity now being used to a large extent for the solution of thermodynamic problems is one which is called *thermal potential*. This is defined as the sum of the energy and the product $Apv$, or

$$I = U + Apv$$

As $U$, $p$ and $v$ are point functions and therefore can be plotted on a set of suitable coordinate axis, it follows that $I$ is a point function and that it can be plotted likewise. Differentiating:

$$di = du + Apdv + Avdp = dq + Avdp$$

Other advantages in the use of $I$ as a coordinate are:

[1] G. A. Goodenough, Principles of Thermodynamics, Henry Holt and Co.

(*a*) If the pressure is constant the heat absorbed (or rejected) is equal to the difference in the value of $I$ at the beginning and end of the process.

(*b*) In a closed cycle the work performed during compression is the difference in the value of $I$ at the beginning and the end of compression.

(*c*) During a throttling process (the process in passing through the expansion valve) the value of $I$ remains constant.

**The *P–I* Diagram.**— The entropy diagram is useful in understanding the manner of heat flow, but it is not workable to the extent of another diagram called the *P–I* diagram (Fig. 1*d*) or the *I–S* diagram (Fig. 1*c*). The *P–I* diagram is drawn out accurately in Figs. 164, 165, 166, 167 and 168, and these may be used to advantage in the solution of engineering problems.

**Perfect Gases.**— The relationship existing between pressure, temperature and volume for a perfect gas may be expressed by the equation

$$pv = MBT$$

or

$$\frac{pv}{T} = \frac{p_1 v_1}{T_1} = \text{a constant}$$

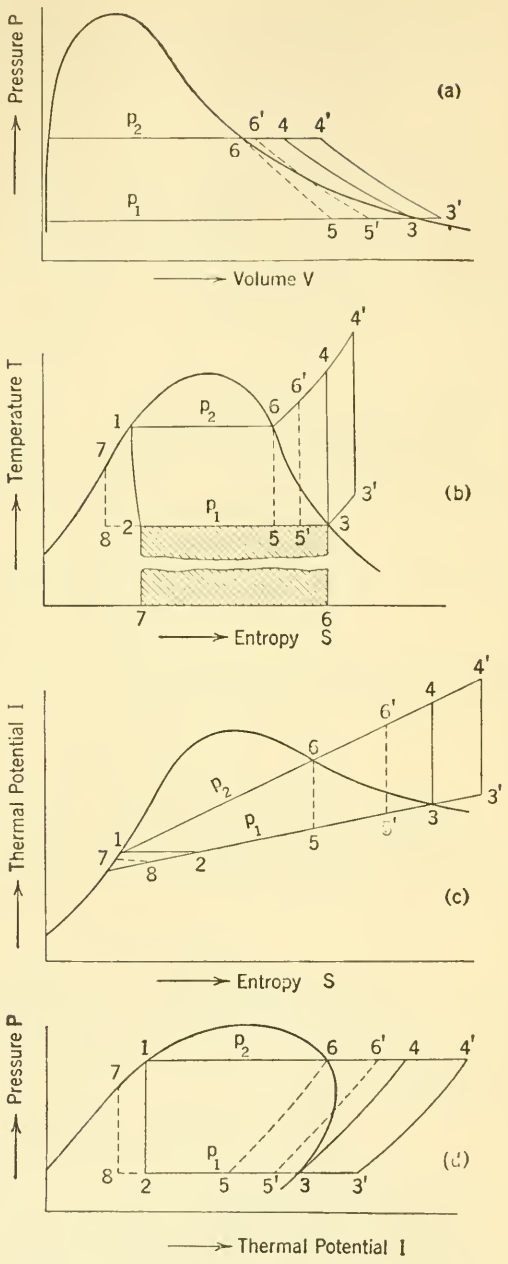



FIG. 1.—Theoretical Refrigerating Cycles for Vapors.

where $p$ is the pressure in pounds per square foot.
$v$ is the volume of the gas in cubic feet.
$M$ is the weight of the gas in pounds.
$B$ is the gas constant, 53.3 for air.
$T$ is the temperature of the gas in degrees F. abs.

**Specific Heat of Gases.**—The thermal capacity of a substance is the heat required to change by unity one of the variable quantities. The *specific heat* of a substance is the ratio of the thermal capacity to the thermal capacity of water. The usual specific heats of gases are those at constant pressure ($c_p$) and constant volume ($c_v$). The ratio of $\frac{c_p}{c_v} = k$ is very nearly constant at 1.4 for diatomic gases, and it is approximately 1.3 for carbon dioxide.

**Intrinsic Energy.**—The intrinsic energy of a perfect gas has been proved (Joule's Law) to be independent of the volume, and dependent only on the temperature of the gas. This relation is expressed by

$$U_2 - U_1 = Mc_v(t_2 - t_1)$$

where $U_2 - U_1$ = the change of intrinsic energy.

$(t_2 - t_1)$ = change of temperature.

This formula may be written also,

$$U_2 - U_1 = \frac{p_2v_2 - p_1v_1}{k - 1}$$

**Changes of State:** *Isothermal.*—Should a perfect gas undergo a change of state at constant temperature, then $pv = p_1v_1$ = a constant and the energy change is zero. From the energy equation

$$Q = AW$$

where $W$ is the work performed in foot-pounds. The projected area under the curve on the $p$-$v$ plane, on the volume axis, gives the external work in every case. For the isothermal the work is given by the expression

$$AW = AMBT_1 \log_e \frac{v_2}{v_1} = Ap_1v_1 \log_e \frac{v_2}{v_1}$$

where $v_2$ and $v_1$ represent the final and initial volumes respectively, and $e$ is the base of the Napierian logarithms.

*Adiabatic.*—During the adiabatic change of state $Q = 0$ and, from the energy equation,

$$U_2 - U_1 = -AW$$

In other words, during the adiabatic change, work is performed at the *expense* of the intrinsic energy or vice versa. The equation for an adiabatic in terms of $p$ and $v$ is given by $pv^k$ = a constant, where $k = \frac{c_p}{c_v}$.

The work performed during such a process is given by

$$ {}_1W_2 = \frac{p_1v_1 - p_2v_2}{k - 1} $$

The relation between the temperatures during an adiabatic process can be shown by

$$ \frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{k-1} = \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} $$

*The Constant Pressure Process.*—During the isobar change of state the heat absorbed is

$$ {}_1Q_2 = Mc_p(t_2 - t_1) = Mc_v(t_2 - t_1) + Ap(v_2 - v_1) $$

*The Polytropic Change.*—In the polytropic change of state the projected curve on the $pv$ plane can be expressed by the equation

$$ pv^n = \text{constant} $$

where $n$ is some exponent which is usually between 1.0 and 1.4 for air The work performed is

$$ {}_1W_2 = \frac{p_1v_1 - p_2v_2}{1 - n} $$

and the energy change (as in any change of state) is

$$ U_1 - U_2 = \frac{p_1v_1 - p_2v_2}{k - 1} $$

**The Air Refrigerating Machine.**—When air or a so-called perfect gas is used for refrigeration it becomes necessary, for economy, to use a compressor and an expansion motor on the same shaft. In such a machine the process for the cooling of commodities will be represented by the line $EA$ (Fig. 2), where $A$ is the temperature of the cold room and $F$ denotes a temperature which is slightly higher than the cooling water

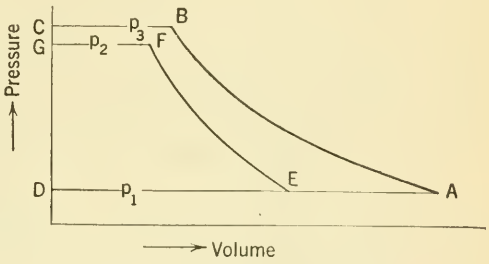



FIG. 2.—Refrigeration Cycle for Air, or a Perfect Gas.

in the aftercooler. The heat absorbed along the line $EA$, assuming complete expansion in the expansion cylinder, is therefore

$$_EQ_A = Mc_p(t_A - t_E).$$

The net work required to be delivered to the air refrigerating machine (not allowing for friction) is the difference in the area $ABCD$ and $EFGD$. The area $ABCD$ is represented by

$$W_{\text{compressor}} = \frac{n}{n-1}(p_1v_A - p_3v_B) = \frac{n}{n-1}MB(T_A - T_B)$$

The area $EFGD$ is expressed by

$$W_{\text{motor}} = \frac{n}{n-1}(p_2v_F - p_1v_E) = \frac{n}{n-1}MB(T_F - T_E)$$

The effect of clearance is to decrease the capacity of the compressor and motor and the result is that the size of both must be increased in proportion. Clearance does not increase the power requirements theoretically but it does practically because of the effect of friction. The formula for the volumetric efficiency is given in Chapter II.

In the formula for the refrigerating effect it is seen that the heat absorbed is directly proportional to the weight $M$. This weight can be increased per cubic foot of piston displacement by increasing the density of the gas during the suction stroke, and this is the characteristic of the so-called dense air machine.

**Vapors.**—Practically all refrigerating machines at the present time use volatile liquids, and it is necessary to understand the thermodynamics of liquids and vapors in order to have a clear understanding of the action of the machine. In each of the diagrams mentioned, the $p$–$v$, the $P$–$I$ and the $I$–$S$ diagrams, there is a liquid line on the left and a dry, saturated line on the right (Fig. 1). The top of the dome made with these two lines is called the *critical temperature*. The region between the liquid and the dry, saturated lines is called the saturated region while the superheated region lies on the extreme right where the vapor begins to show characteristics of a perfect gas.

During the absorption of heat by the liquid the temperature will rise and the volume will increase very slightly. The heat of the liquid (if the specific heat is taken as a constant) is given for one pound of the substance by:

$$_1Q_2 = c \times (t_2 - t_1)$$

and if the specific heat is not constant

$$dq = cdt$$

where $(t_2 - t_1)$ is the increase in the temperature during the addition of the heat.

If the value of $c$ is not constant then the method of summation must be employed, namely,

$$Q = \int c dt$$

If, at any particular temperature, the action of evaporation takes place heat will be absorbed at constant temperature (and constant pressure) until the liquid is entirely evaporated, and the amount of heat so absorbed per pound is called the *latent heat* of vaporization ($r$). If after the liquid is entirely evaporated any further heat is absorbed the substance becomes *superheated* and (if the pressure is maintained constant) this occurs at an increasing temperature. The heat absorbed in the superheated region is similar to the action of increasing the temperature of the liquid, and for one pound of the substance, still keeping the pressure constant

$$_1Q_2 = c_p \times (t_2 - t_1)$$

if the specific heat were constant, but as the value varies, then, as before the heat absorbed becomes

$$Q = \int c_p dt$$

There is no standard starting point for the measurement of heat content. For a long time the zero point has been taken as 32 deg. F., but this is awkward in refrigeration so that the Bureau of Standards used $-40$ deg. F. as their zero in the tables of the properties of saturated and superheated ammonia. At this initial state the thermal potential of the liquid becomes

$$i'_0 = u_0' + Apv_0$$

As $u'_0 = 0$ at this point,

then $i'_0 = Apv_0$

where $u_0$ and $v_0$ are the intrinsic energy and volume of one pound of the liquid at the zero of the measurement of heat content.

For example, taking $p = 144 \times 10.41$ lb. per sq. ft.

and $$v_0 = \frac{1}{43.08},$$

then $$i'_0 = \frac{144 \times 10.41}{778 \times 43.08} = 0.0448 \text{ B.t.u.}$$

The significance of $i'_0$ can be seen by considering the work performed in forcing one pound of liquid into the container against the pressure $p$ lb.

The value of $i$ is given by the expression $i = i' + xr = i'' - (1.0 - x)r$ for saturated vapor of a quality $x$, where the term $x$ represents the decimal part (of the unit pound) that is a vapor.

The volume of a mixture is, approximately, $v = xv''$ where $v''$ is the volume of 1.0 lb. of dry, saturated vapor at the pressure under consideration.

In the tables of the properties of saturated vapors may be found the pressure, temperature, volume of one pound of saturated vapor, thermal potential of the liquid and of the dry saturated vapor, the latent heat and the entropy of the liquid and of saturated gas. If the superheated region is evaluated, the specific volume, the value of $I$ and the entropy are given for the different pressures and amounts of superheat.

**The Refrigerating Cycle.**—The refrigerating cycle for volatile liquids is shown in Figs. 1*a*, 1*b* and 1*c*. Point 1 represents the liquid under a pressure of $p_2$ lb. corresponding to some liquefaction temperature usually between 70 and 90 deg. F. In order to perform useful refrigeration the pressure must be reduced to such an amount that the corresponding boiling temperature will be suitable for the conditions prevailing, say, 10 to 20 deg. F. The action of the expansion valve (the pressure reducing valve) is shown by the line *12*. Absorption of heat by the refrigerant takes place along the lines *23* and on the *T*–*S* plane the amount of heat absorbed is given by the *area 2367*, but on the *I*–*S* diagram the heat absorbed is given by the difference $(i_3 - i_2)$. If the gas with a condition represented by *3* passes to the compressor, superheated gas enters the condenser as the compression line is given by *34*, and the point *4* represents a condition of superheat. If the compression line were represented by the line *56* the condition *5* would indicate the presence of a liquid returning with the vapor, the amount of liquid in this case being sufficient for the point *6* to be just on the dry saturated vapor line. If a greater amount of liquid returned with the vapor then the point *6* might be farther to the left and this would indicate that even after compression there would remain some liquid in the compressed vapor. Finally, the compressed, superheated gas enters the condenser and is cooled and liquefied through the absorption of heat by the condensing water along the line *461*. On the *I*–*S* diagram the useful refrigeration is the value $i_3 - i_2$, the work of compression is $i_4 - i_3$ and the heat lost to the condenser water is $i_4 - i_1$. The tables and the diagrams are usually calculated for 1.0 lb. of the refrigerant.

The object of the compressor is simply to increase the pressure as efficiently as possible to an amount sufficiently great so that the refrigerant may be condensed into the condition of a liquid again by the

use of atmospheric air or, in nearly every case, by the use of well or surface water, thereby making it possible to use the same refrigerant continuously. The actual discharge pressure of the compressor, therefore, depends on the temperature and the amount of the cooling water showered over the condenser. The function of the condenser is to liquefy the refrigerant, but it is very important that the pressure of liquefaction be as low as possible as well as the temperature of the liquid at the expansion valve. This can be seen very well from the *I–S* diagram.

The refrigerating effect (from Fig. 1*c*) is seen to be $i_3 - i_2$ B.t.u. per pound, where $i_2 = i_1$, the thermal potential of the liquid at the expansion valve. The lower this value is the greater will be the refrigerating effect $i_3 - i_2$, but the only economical manner of lowering this temperature is by the use of cooling water over the condenser or by the use of a water-cooled aftercooler. During the compression process, if the compression line can be represented by the expression $pv^n$ = a constant, the temperature rise may be excessive and is given by the formula

$$t_2 = (t_1 + 460)\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 460$$

where $t_2$ = the final discharge temperature in deg. F., $t_1$ = the initial temperature in deg. F., and $p_2$ and $p_1$ are the final and the initial pressures respectively.

If recourse is made to the water jacket only in order to keep from overheating the cylinder the temperature of the discharge will become high at times, sometimes as great as 300 deg. F. At one time it was generally understood that improved operation could be obtained by the use of a liquid refrigerant injection into the suction of the compressor and such a process has the term *wet compression*. Such a means of operation is still used in extreme conditions but the control is difficult and few engineers advocate wet compression at the present time for general practice.

**The Ton of Refrigeration.**—The American ton of refrigeration is the cooling rate of 200 B.t.u. per minute or 12,000 B.t.u. per hour. The standard operating conditions are an evaporation temperature in the cooling coils of 5 deg. F. and a liquefaction temperature in the condenser of 86 degrees. The unit of refrigeration in British practice is the cooling effect of one kilogram calorie per second. This would be $3.96 \times 60 = 237.6$ B.t.u. per minute or about 18.8 per cent greater than the American unit. The American ton of refrigeration got its origin from the consideration of the melting effect of 2000 lb. of ice in

24 hr. with the latent heat of fusion, taken in round numbers, as 144 B.t.u. per pound.

**The Coefficient of Performance.**—According to the laws of thermodynamics the work performed in compressing different refrigerants for the same heat absorbed from the refrigerator is the same (this being another way of expressing the second law) for similar sets of conditions. In practice there are variations which assume relative proportions of as much as 20 per cent for carbon dioxide as compared with ammonia. This variation is due to the relatively large value of the heat of the liquid as compared with the latent heat of vaporization and because a pressure reducing valve is used instead of an expansion cylinder. It is important that the power consumed be known for the different refrigerants under varying operating conditions. This amount can be found, among other ways, from the ratio of the refrigerating effect to the work of compression. On the *I–S* diagram, Fig. 1*c*, the ratio would be $\frac{i_3 - i_2}{i_4 - i_3}$. For efficient work the value of the coefficient of performance should be large, with theoretical values of from 4 to 10, but in practice the coefficient is considerably less.

**Horse Power per Ton of Refrigeration.**—In American practice it is more usual to hear the term horse power per ton of refrigeration than the coefficient of performance. Using again the *I–S* diagram such a value would be obtained theoretically from

$$\text{horse power per ton} = \frac{i_4 - i_3}{i_3 - i_2} \times \frac{200}{42.44} = 4.713 \frac{i_4 - i_3}{i_3 - i_2}$$

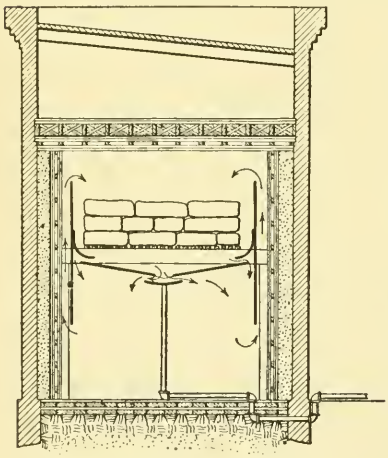


Fig. 3.—Refrigeration by the Use of Ice.

In practical work the real volumetric efficiency would need to be considered, and this would increase the values given in the formula. Values for horse power per ton of refrigeration may be less than 1.0 up to 2.0 or more depending on the operating conditions.

**Refrigeration.**—Refrigeration, as will be noted later, involves a multitude of different applications. In general it is understood to embrace any method of securing a temperature lower than that of the atmosphere. If this is done by means of ice and salt, or other mix-

tures, or by the use of ice alone (Fig. 3) the work is limited and expensive. The temperatures produced by the use of various mixtures are as follows:

TABLE 1

| | Parts by Weight | Degrees F. | | Parts by Weight | Degrees F. |
|---|---|---|---|---|---|
| Sodium chloride | 1 | −1.6 | Sodium chloride | 1 | −0.4 |
| Snow (or ice) | 3 | | Snow | 1 | |
| Calcium chloride | 1½ | −27.4 | Calcium chloride | 2 | −43.6 |
| Snow | 1 | | Snow | 1 | |
| Dilute nitric acid | 1 | −31.0 | Saltpeter | 1 | |
| Snow | 1 | | Sal ammoniac | 1 | −11.2 |
| | | | Water | 1 | |
| Potassium hydrate | 4 | −38.3 | Sulphuric acid | | |
| Snow | 3 | | Nitric acid | 1 | −40.0 |
| | | | Snow | 2 | |

Such a method of securing refrigeration is cumbersome, limited as to the temperatures secured and the ability to secure ice and the cost of producing the cooling would be too great for general use. If the cooling is done by means of a machine of any sort, the operation is called *mechanical refrigeration.*

**The Simple Refrigerating Cycle.**—The modern refrigerating plant, Fig. 4, uses a closed cycle, even when air [2] is used as the refrigerant. The complete cycle includes then: a set of refrigerating coils through which the refrigerant circulates and absorbs heat from the cold body; an aftercooler or condenser (at the upper pressure) which is the hot body and which receives heat; and a reversed engine, Fig. 5, usually called the compressor. Between the condenser and the refrigerating coils is the throttle valve for reducing the pressure to the amount best suited for the conditions in the refrigerating coils. A well-designed plant must have the work of each of these three parts of the cycle well

[2] A closed cycle makes it possible to operate with suction pressures above one atmosphere as in the case of the dense air machine. With such closed cycles the moisture in the air may be removed so as to decrease the operating troubles due to frost accumulations on the expansion engine valves and exhaust ports. It so happens that in the early days of mechanical refrigeration the open system was used more extensively than the closed cycle.

balanced. The refrigerating coils must be large enough in heat transfer surface to absorb the required amount of heat from the substance to be cooled, for example, air, liquids, etc., in order that the required " duty " shall be performed. The absorption of heat involves consideration of the value of heat transfer usually stated as the number of B.t.u. per

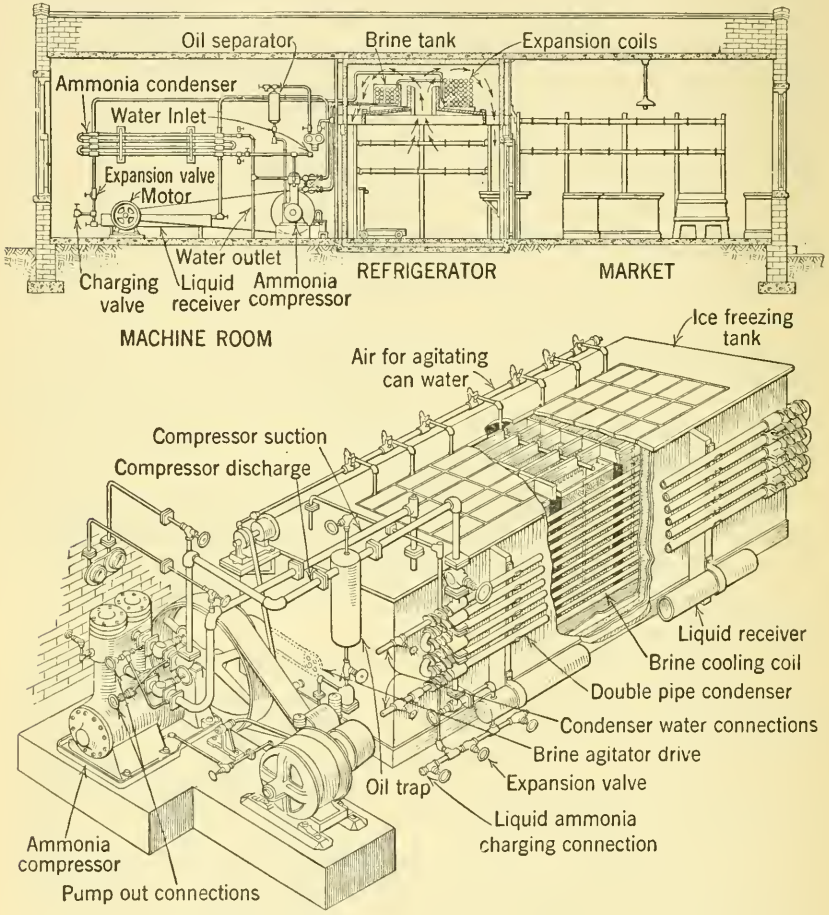
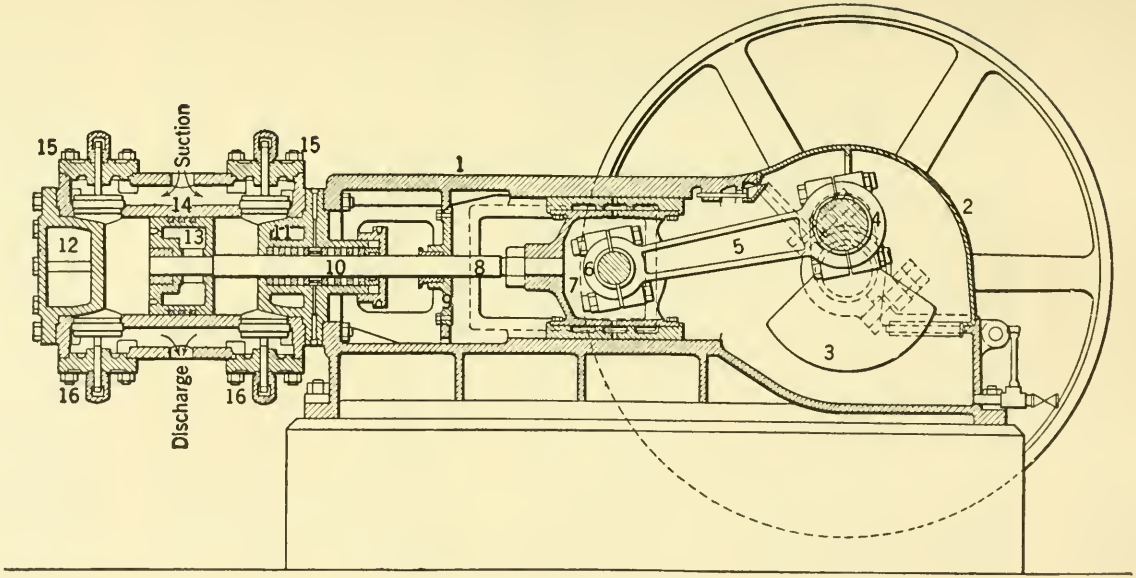



Fig. 4.—Typical Refrigeration Plants.

square foot per 1.0 degree F. difference of temperature per hour and is represented by the symbol $k$. This value $k$ is of extreme importance in mechanical refrigeration.

The aftercooler (the condenser) has to remove a certain amount of heat, an amount varying from 230 to 250 B.t.u. per minute per ton of




SECTIONAL ILLUSTRATION

1. Frame
2. Crankcase Cover
3. Counter Balanced Crankshaft
4. Crankpin Bearing
5. Connecting Rod
6. Crosshead Pin Bearing
7. Crosshead
8. Piston Rod
9. Oil Guard with Wiper Rings
10. Stuffing Box for Piston Rod
11. Front Cylinder Head
12. Back Cylinder Head
13. Piston
14. Cylinder
15. Suction Valve
16. Discharge Valve

FIG. 5.—Typical Horizontal Double Acting Ammonia Compression Cylinder.

refrigeration. Again the value of the coefficient of heat transfer ($k$) is an important consideration and in each case the total heat transfer may be stated from the formula:

$$Q = A \times k \times (t_2 - t_1)$$

where $A$ = the area of the surface exposed to heat transfer;
$t_2 - t_1$ = the temperature difference on the two sides of the surface;
$Q$ = the heat absorbed or rejected per hour.

**Expansion Cylinder or Expansion Valve.**—As the energy change of a perfect gas is a function of the temperature only it becomes essential that when a so-called perfect gas is used for refrigeration that an expansion cylinder be employed in order that on expansion it will at least approximate the adiabatic. In the case of vapors this is different and in practice all successful refrigerating machines using a liquid refrigerant make use of the throttle (pressure reducing) or the so-called " expansion " valve. The gain in simplicity, cheapness in first cost and ease in operation as compared with some slight gain in economy when using the expansion cylinder has entirely excluded the latter. As an example, take a condition of 86 deg. F. temperature of the liquid ammonia and 5 deg. F. evaporating temperature. In the expansion valve $i_1 = i_2$ or

$$i'_{86} = i'_5 - x_5 r_5 \quad \text{or} \quad x = \frac{138.9 - 48.3}{565.0} = 0.1605$$

whereas in the adiabatic expansion (constant entropy)

$$s'_{86} = s'_5 + x_2 \left(\frac{r}{T}\right)_5 \qquad x = \frac{0.2875 - 0.1092}{1.2161} = 0.1470$$

From this calculation the loss of refrigeration due to the use of the expansion valve is $(0.1605 - 0.1470)\ 565 = 7.6$ B.t.u.

**Sub-cooling the Liquid.**—The refrigerating effect, per pound of ammonia, boiled to dry, saturated vapor with $t_1$ before and $t_2$ after the expansion value is

$$Q = i''_2 - i'_1$$

It is evident that any scheme which will decrease the value of $i'_1$ (the thermal potential of the liquid before it enters the expansion valve) will result in a greater value of the net available refrigeration per pound of refrigerant, but in general the operation must be done by the condensing water. However if colder water than that used for liquefaction in the condenser is available it must be used for sub-cooling *after* liquefaction as, if any vapor is present sub-cooling is not possible unless air or other non-condensible gases are present as is shown in Chapter IV.

The only other successful method of sub-cooling the liquid is to make use of the dual compression or the stage compression method given in Chapter II whereby the liquid is cooled at an intermediate pressure and the gas evolved at the intermediate pressure is compressed by a special compressor.

**The Optimum Refrigerant.**—The second law of thermodynamics states that the kind of working medium does not influence the economy of operation of a heat engine. A preferred refrigerant, therefore, will not be one that will increase the efficiency of the cycle, but it will be one that for the kind of operation, or the type of load or service will give best results as regards the pressures exerted in the cycle, the piston displacement, the chemical properties or the effects on plant or animal life. Some of the small machines are required, by reasons of convenience, to use an air-cooled condenser in which case the pressure resulting from the liquefaction must not be excessive.

In the following chapters of this handbook, then, there will be considered first the compressor or other mechanical device between the lower (the refrigerating) pressure and the higher (the condenser) pressure, Chapter II; the absorption machine, Chapter III; condensers, fittings, etc., Chapter IV; the low-pressure side and its applications to various industries; as for example that of ice making, Chapter XIII; cold storage, Chapter XIV; and miscellaneous applications, Chapter XV. Subjects of lesser general importance, though valuable to the engineer, such as heat transfer, Chapter VI; automatic refrigeration, Chapter V; brine, Chapter VIII; water-supply systems, Chapter IX; operation and erection, Chapter X, etc., are discussed as indicated.

## CHAPTER II

## THE COMPRESSOR

**History and Development.**—The first machines used in the mechanical production of cold were constructed on the principle of the vacuum machine, wherein the vacuum—which was obtained mechanically—permitted the refrigerant to boil at a sufficiently low temperature to secure the results desired. These machines included that of Wm. Cullen in 1755, using water under high vacuum; Vallance's machine of 1824; and Edmund Carré's device of 1850, both using sulphuric acid to absorb the water vapor, and thereby permitting less severe operating conditions than would be required if an air pump had had to be used.

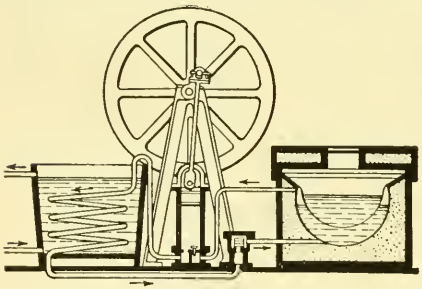


Fig. 6.—Vapor Compression Refrigerating Machine of Jacob Perkins.

In 1845 Dr. John Gorrie developed the cold air machine, using a closed cycle and an expansion cylinder, and this machine was improved by Kirk in 1861, Postle in 1868 and Windhausen and Nehrlich in 1869. The air machine was perfected by James Coleman and John and Henry Bell in 1877 and subsequent years, with the result that the Bell-Coleman compressor became very well known and mechanical refrigeration got a real impetus. The present-day compressor is based on the designs of Linde, Ferguson and John De la Vergne.

The first refrigerant other than air or water was that used by Jacob Perkins whose compression machine (Fig. 6) was invented in 1834. This machine was intended to be used with sulphuric ether, $[(C_2H_5)_2O]$ but it never got beyond the experimental stage. The Perkins machine was revived in 1857 by Dr. James Harrison (using sulphuric ether) and was quite successful for refrigeration in breweries, and for the cooling of meat and other perishable products. Finally Dr. Carl Linde, in 1873, introduced the ammonia compression machine, and in 1876

Raoul Pictet devised the sulphur dioxide compressor, using water to cool the piston and piston rod as well as the cylinder. In the eighties mixtures[1] of different refrigerants, and volatile liquids like ethyl chloride, methyl chloride, ethylic ether, acetylene, ethylo-sulphurous dioxide, naphtha, gasoline, cyrogene, etc., were given attention. By a system of elimination the practical refrigerants were narrowed—as far as Great Britain and the United States were concerned—to ammonia and carbon dioxide, but at the present time (1927) interest is directed towards other refrigerants for special purposes.

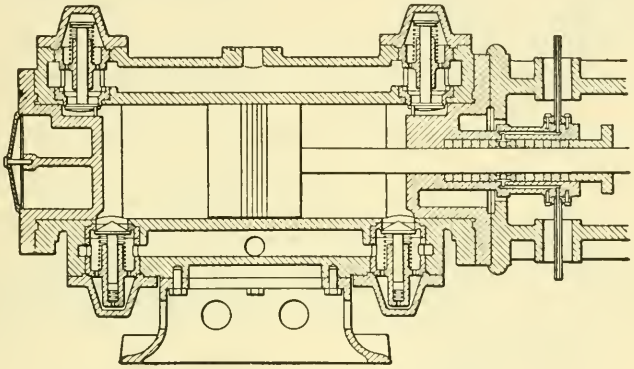


Fig. 7.—Apeldoorn Compressor.

## THE MODERN DESIGN OF AMMONIA COMPRESSORS

**The Slow-speed Horizontal Double-acting Compressor.**—The present-day construction of the older type (the Linde design) of ammonia compressor of the double-acting, horizontal design (which is still preferred by certain engineers) is heavy and sturdy. The design is more like the heavy-duty rolling-mill engine, and has sufficient metal in the cylinder for several reborings. The frames are usually made of very close-grained cast iron, while the cylinder is either of a close-grained cast iron or of a special steel casting and is cast separately from the frame. The cylinder heads are made of the same material as the cylinder, and the Linde type of compressor has a spherical piston and

[1] A number of mixtures of two vapors has been attempted of which Pictet's liquid is the best known. This latter is a certain mixture of carbon dioxide and sulphur dioxide and the resulting combination condenses at a pressure considerably less than required for pure carbon dioxide. More recently propane and butane have been proposed but they are mere mixtures and not new compounds. The physical properties of such *mixtures* are never known with any degree of accuracy, and the relative amounts of the mixture vary considerably unless it is possible to eliminate all leaks.

cylinder heads, the idea being that greater valve area as well as greater strength is possible by this arrangement.

In this type the piston is now, usually, of semi-steel, cast in one piece and hollow, although the built-up construction using the spider, spring and bull ring is also common. The crosshead is usually of the box type with adjustable shoes at top and bottom, and is of semi-steel. The connecting rod is made of one-piece steel forging, with bronze boxes, lined with babbitt. The main bearing is of the floating type of quarter box, lined with babbitt, and is adjusted by means of wedges and screws (Fig. 8). The crankshaft is of one-piece steel forging, turned and ground. The stuffing box is of the lantern type, using now, as a rule, first a metallic packing; then the lantern, and finally, a soft packing. The lantern is connected by means of piping to the suction line so that there is in this way only suction pressure exerted on the outer part of the stuffing box, whose length may vary in the different makes from 8 to 18 in. (Fig. 9).

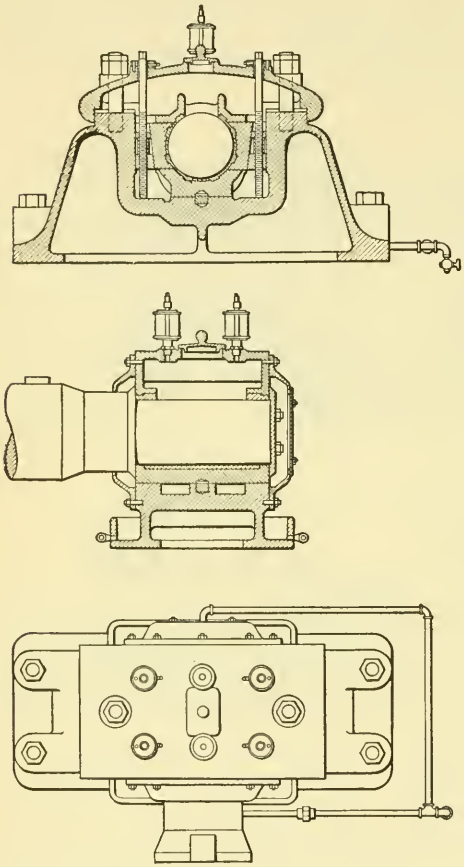


Fig. 8.—Compressor Details; the Main Bearing.

The older Linde type of valve is the poppet, or mushroom valve which gives " line contact " on a bevel seat. The poppet valve has been remarkably successful. Springs are used to close the valves, but an attempt is made to cushion the closing of the valve (which is usually quite heavy) by means of dash pots (Fig. 10). On account of the sturdy design and workable valves the compressors of the Linde type have had a very successful life, the design not having been changed to any great extent since the first construction in about the year 1873.

The poppet valve can be made very tight, and it can be reground on its seat easily. The compressor speed is limited, however, to from 75 to 100 r.p.m., and therefore where higher speeds are required a lighter weight valve—one of small inertia—must be used.

**The Horizontal High-speed Compressor.**—The principal reason for the use of the high-speed in place of the low-speed compressor is the decrease in the space required by the machine, although a cheaper electric motor can be used if it is permitted to revolve at 1200 or 1800 r.p.m., and the oil engine drive is easier to make at compressor speeds up to 250 r.p.m. It is a mistake to say, however, that the moderate-speed compressor is cheaper per ton of refrigeration, as the stresses developed make it necessary to use higher-priced material than would

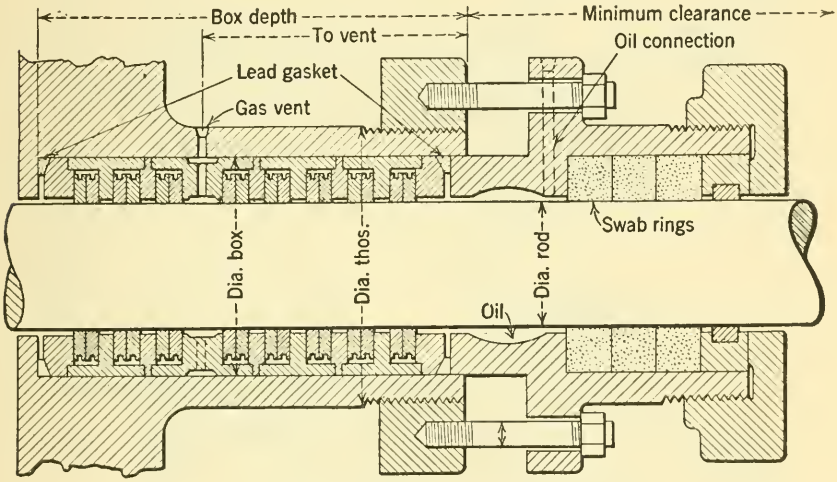



Fig. 9.—Metallic Rod Packing.

be used in the slow-speed machine, and the problems of balancing, lubrication and foundations demand special consideration, in consequence of which the cost is considerably increased, as compared with the slower rotative speeds using the same sized cylinders.

The high-speed compressor frame is of the rolling-mill engine type, and the frame, cylinder and heads are composed of semi-steel (Fig. 11). The castings are sand blasted and then have an acid bath to remove the surface sand. The piston and connecting rods and the crankshaft are of high quality steel forgings, each forging being made from a single billet. The pins are of high carbon steel, heat treated and ground. The valves are of the plate (the De la Vergne, and Ingersoll-Rand), Fig. (11) the ring plate (Vilter, and discharge valve of the Arctic) and

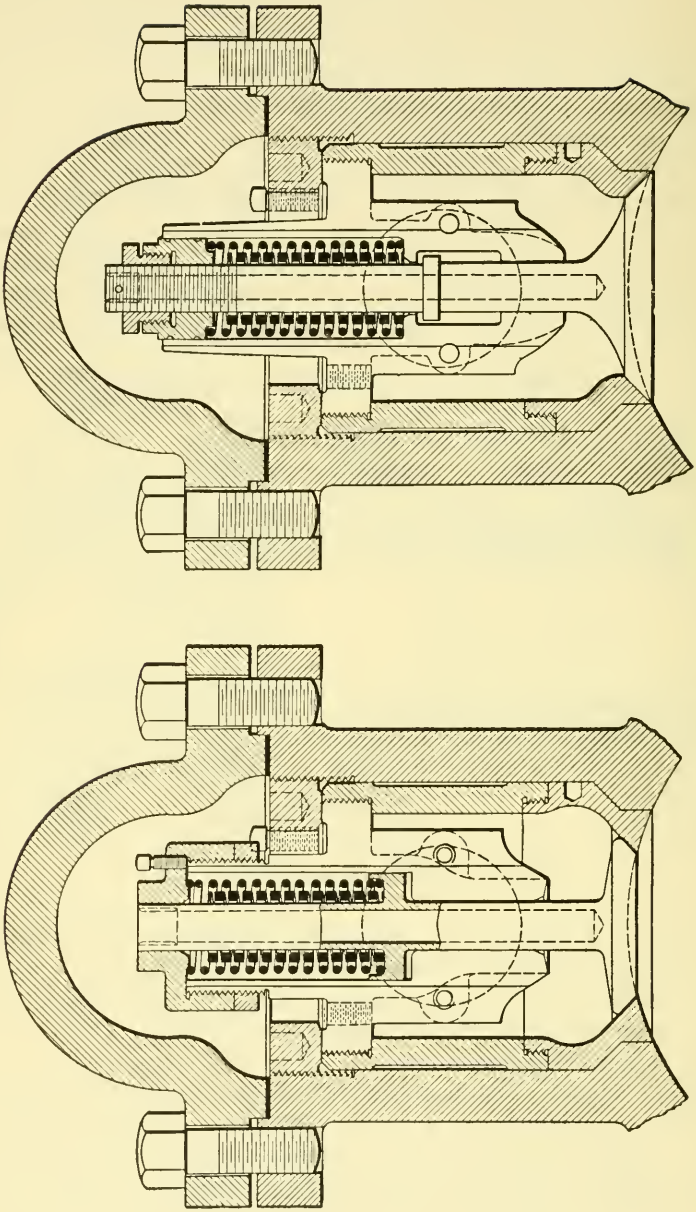


Fig. 10.—Compressor Details; Poppet Valves.

the ribbon (the Carbondale) type. The material composing the valves is the very best, frequently chrome-vanadium steel, and is hardened and ground, or tempered and ground. The rise of the valve off its seat is very slight. Typical indicator cards are shown in Figs. 13 and 14.

In the horizontal machines the discharge valves are always placed on the bottom in order to permit easy relief should liquid ammonia enter the cylinder with the suction gas. The high-speed compressor uses sets of valves of small size in preference to a smaller number of large valves. The suction and the discharge valves are frequently identical, except for the position of the valve cage in the seat. The ports are

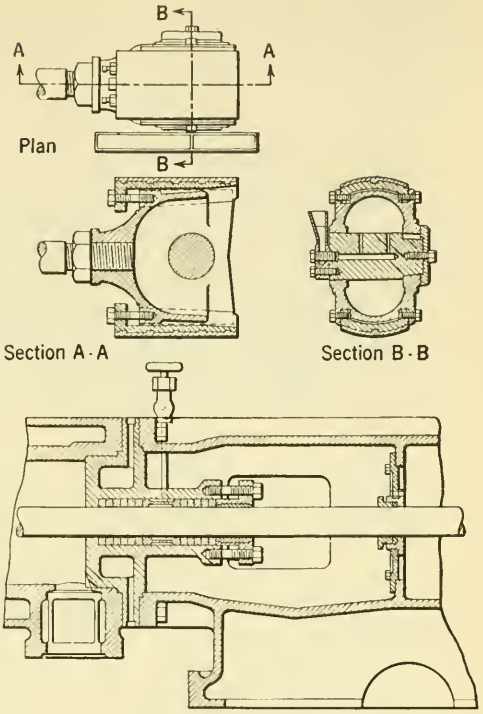



FIG. 11*a*.—Compressor Details; Crosshead and Rod.

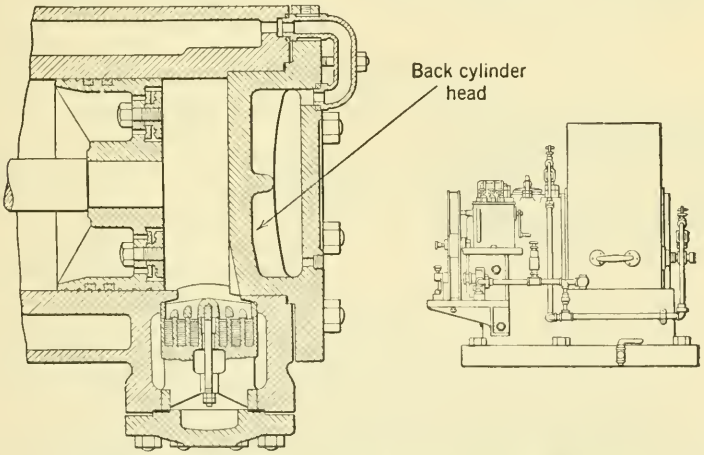



FIG. 11*b*.—Compressor Details; Head End and Valves.

made generous in size and as direct as possible. Tests indicate that the port design affects the economical operation to a greater degree

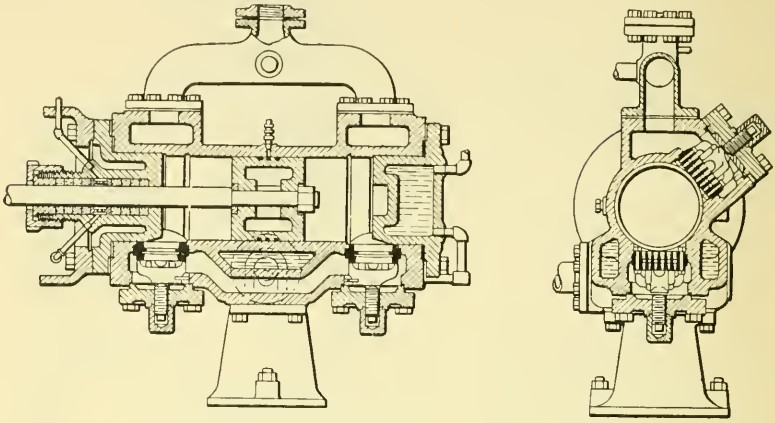


FIG. 12.—Horizontal Double-acting Ammonia Compressor with Plate Valves.

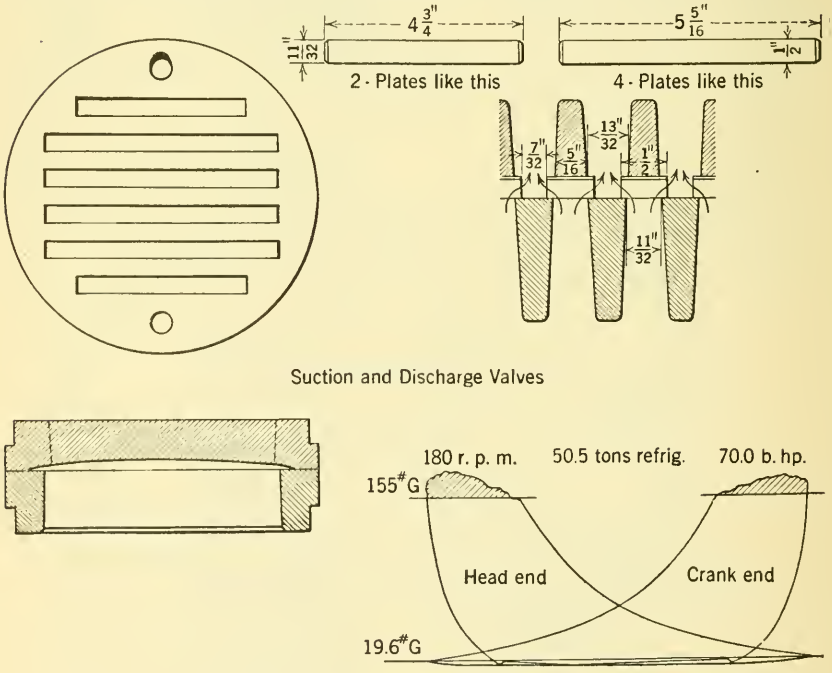



FIG. 13.—Details of the Ribbon Type of Valve, with Indicator Card.

than the net cross section of the valve opening. Since designers act with the understanding that a nominal amount of clearance—varying

from 3 to 6 per cent—does not affect appreciably the economical operation of refrigerating compressors, the horizontal, double-acting high-speed compressor usually does not attempt to eliminate clearance, or attempt to utilize the safety head principle, nor is there an attempt to safeguard the cylinder, by flexible cylinder head studs or otherwise, from the harmful effects of slugs of ammonia liquid in the cylinder.

Special consideration, however, is given to *lubrication*. There are now two methods in frequent use; first, the sight feed automatic lubricator (the Vilter and the De la Vergne) and second, the splash type for the engine crank mechanism (the Ingersoll-Rand, the Arctic and the Worthington). When the latter is used, provision is required in the shape of a special partition to prevent the oil in the crankshaft from entering the cylinder. Cylinder (low temperature test) oil is pumped into the cylinder and the stuffing box by means of a special mechanical pump lubricator.

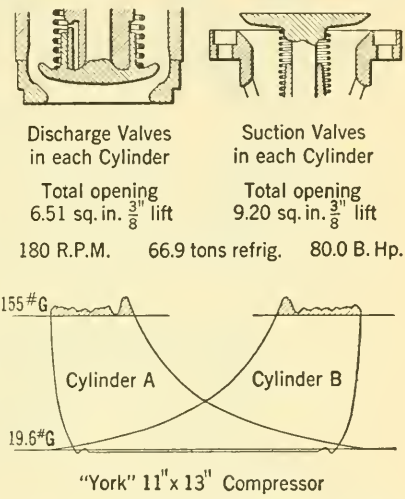



Fig. 14.—Details of the Poppet Type of Valve, with Indicator Diagram.

As it is practically impossible so to control wet compression as to obtain the best results, all modern compressors operate on dry compression, thus necessitating (for ammonia) the use of a water jacket. The water jacket is of slight use except at the last third of the stroke, and so the heads are practically the only place where the horizontal, double-acting compressor can obtain any benefit. This tendency is seen in the Vilter, the Ingersoll-Rand and the Worthington Pump designs (Figs. 11, 12 and 24). It is a question whether the water jacket affects the compression line appreciably, but its function is to prevent overheating of the metal, and in consequence to improve at the same time the " real " volumetric efficiency.

**The Vertical Single-acting Compressor.**—The vertical single-acting compressor is of two types, (1) the semi-enclosed type and (2) the totally enclosed designs. The semi-enclosed twin cylinder open frame compressor first appeared about 1900 and is still being sold by the Frick and the York Companies (Fig. 26). It is an excellent design, giving the greatest economy of any compressor, and is built in the

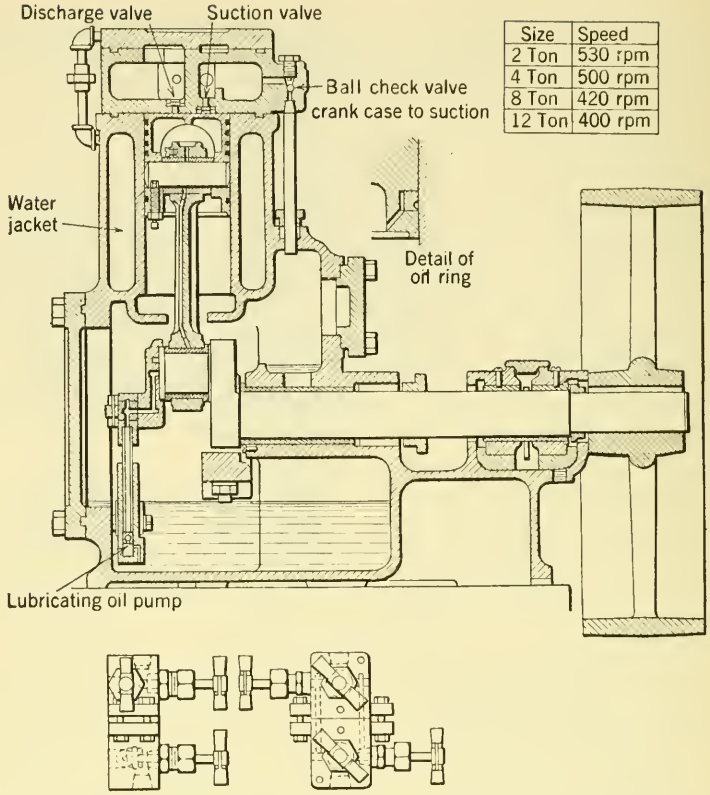



Fig. 15.—Single-acting, Enclosed Type, High-speed Ammonia Compressor.

largest capacities. Because of the vertical cylinder and the use of a crosshead, only piston ring pressure is exerted on the cylinder walls, and the cylinder does not wear out of round as quickly as the horizontal design does. The height of the compressors, the operating troubles with the stuffing box and the relatively slow speed appear to influence opinion in favor of the totally enclosed compressor (Fig. 16). This latter design is becoming more and more liked, both in America and Great Britain. Both types have a uni-flow of the ammonia as the suction ammonia gas enters below the piston and passes through the suction

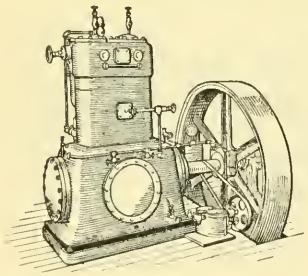


Fig. 16.—Twin Vertical Single-acting Ammonia Compressor.

valve in the piston of the ring plate or the balanced light weight poppet type, the latter opening and closing by the inertia of the valve during reciprocation, and the discharge passing through a form of poppet valve in the false head held on its seat by means of heavy springs (Fig. 14). The water jacket is located along the *last third* of the stroke only, and at times in the head as well, the remainder of the barrel being insulated. On account of the safety false head the (striking) clearance

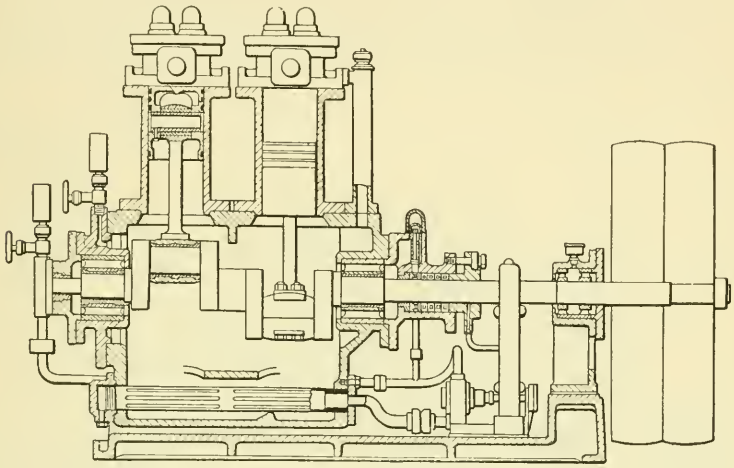


FIG. 17.—The Haslam Ammonia Compressor.

From The Haslam Foundry & Engineering Co., Ltd., Derby, England.

| | Number of Machine | | | | |
|---|---|---|---|---|---|
| | 2 | 3 | 4 | 5 | 6 |
| Heat units eliminated hourly, cooling 60° to 40° F. | 32,000 | 72,000 | 112,000 | 152,000 | 240,000 |
| Ice-making capacity per 24 hours, tons | 1 | 2½ | 3⅔ | 5 | 8 |
| Refrigerating capacity per 24 hours, tons | 2 | 4½ | 7 | 9½ | 15 |
| Actual horse power required | 4½ | 9 | 13 | 16 | 22 |
| Revolutions per minute | 200 | 170 | 150 | 140 | 130 |
| Shipping weight, cwts. | 13¾ | 19½ | 25½ | 36½ | 48½ |
| Shipping measurements, cubic feet | 36 | 75 | 100 | 170 | 220 |

is made very small, as little as $\frac{1}{64}$ in. The piston rods and stuffing box are subjected to suction pressure only.

**The Semi-enclosed Compressor.**—The semi-enclosed vertical single-acting compressor (Fig. 26) has its two cranks 180 degrees apart, and (if engine driven) has the connecting rod of the steam engine connected to one of the crankpins of the compressor. This permits the maximum resistance of the compressor to occur at the same time as the maximum

turning effort of the horizontal engine, and thereby permits a smaller flywheel than would otherwise be needed. A regular crosshead is used, but the stuffing box, being subjected to suction pressure only, need not be as long as is required for the double-acting machine. This stuffing-box packing has hard usage, and as it is very hard to keep the operating conditions constant, and as the packing is likely to become frozen from time to time when liquid ammonia returns with the suction back to the compressor, the liquid is likely to run down the rod. The materials used in the cylinders are close-grained, high-grade, cast iron, but the frame and the remainder of the machine are usually of ordinary material.

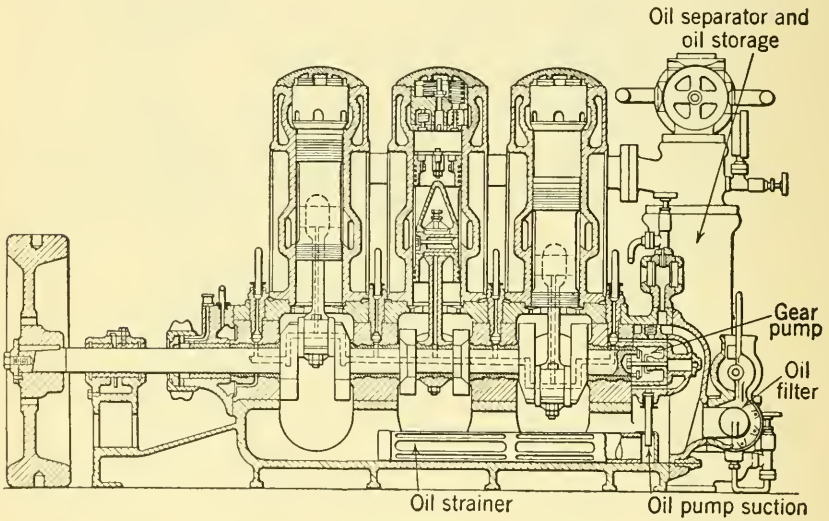



Fig. 18.—The Stearns Ammonia Compressor.

**The Enclosed Compressor.**—The enclosed crankcase compressor (Figs. 15, 16, 18) lends itself best to high speed, or moderate speeds in the larger sizes, to automatic operation and to decreased loss of ammonia. It has been popular in the smaller sizes, say from the $3'' \times 3''$ to the $9'' \times 9''$ but now the enclosed compressor is found in sizes up to $18'' \times 20''$. The feature tending to decrease the loss of ammonia is the fact that the packing and the rod are kept at nearly constant temperature and that the rod has rotation and not translation. A scored rod need not leak under conditions of rotation.

The cylinder construction as regards materials, heat treatment and machine work is practically the same as for the high-speed horizontal compressor. The cylinder is of semi-steel and the crankcase and the

cylinder are sand-blasted and acid-dipped in order to remove any surface sand. The piston is known as the double trunk pattern as it acts in the capacity of a crosshead, and takes the side thrust due to the obliquity of the connecting rod, as well as performing the ordinary function of the piston. The piston in the smaller sizes, up to the $9'' \times 9''$ size, usually has only one balanced suction valve in the piston, but the larger sizes have three or more such valves. The discharge valves, usually of the modified plate or the poppet type of valve, may be three or more in number.

Unusual care is necessary in the design and the construction of this type of compressor. It is subjected to very heavy forces. The machine work in the cylinder and the crankcase must be excellent, and heat treatment usually is necessary in order to prevent warping of the casting after the finishing cuts are made, a consideration that is even more important where three or more cylinders are used (Fig. 19). The crankshaft needs to be very massive and of the best material, particularly when wear on the part of the center bearings means a much longer span than the difference between cylinders. The York Manufacturing Company specified open hearth steel forgings of 75,000 lb. per square inch ultimate tensile strength, and an elastic limit of 40,000 lb. per square inch for both crankshaft and connecting rods.

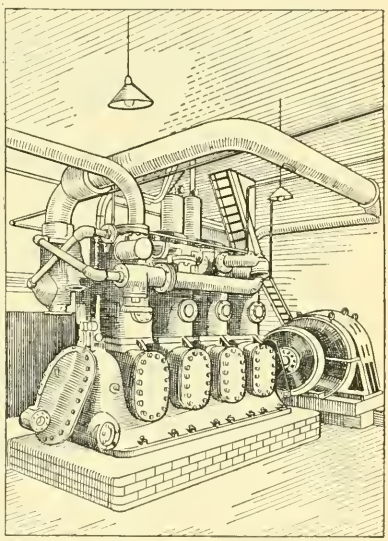


Fig. 19.—The Stearns Compressor.

**Clearance.**—The feeling has prevailed for years that clearance, in an ammonia compressor particularly, is a very bad thing, and for some time past an attempt has been made to reduce the clearance as much as possible. It is now understood that clearance will reduce the capacity of the refrigerating machine, since the gas in the clearance volume will expand at the beginning of the suction stroke until the pressure, having finally been reduced in the cylinder to an amount equal to or less than the pressure in the suction bends, will cause the gas to enter the cylinder and the new *effective* suction stroke will begin. The work performed in the compression of the gas in the clearance volume is very nearly all returned

in the expansion taking place at the beginning of the suction stroke. The loss of capacity, however, is a real one and may reach a large amount, depending on the range of the pressure and the clearance in the cylinder in per cent of the piston displacement. It so happens that certain forms of valves make it necessary to use some 4 or more per cent of clearance, but in such cases the usual procedure is to increase the diameter slightly in order to compensate for the loss of capacity.

**Volumetric Efficiency.**—In refrigeration, when the volumetric efficiency is spoken of, reference is not being made to the *apparent*[2] volu-

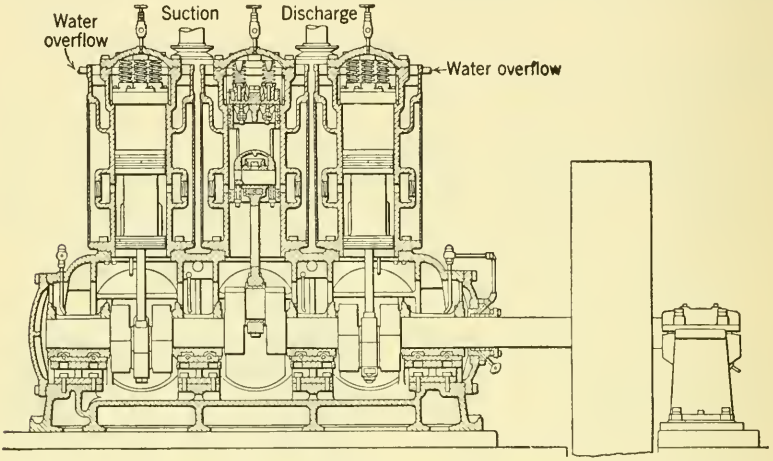



FIG. 20.—The York Ammonia Compressor.

metric efficiency which is caused by the clearance, but to the *real* volumetric efficiency. This is in reality a quantity obtained by dividing

[2] The apparent volumetric efficiency may be found from the following values, using the quantities given in Fig. 21.

$$V_1 = V_p + V_c - V_d$$

$$V_d = V_c\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} \qquad \text{and let} \qquad \frac{V_c}{V_p} = m$$

$$V_1 = V_p + mV_p - mV_p\left[\frac{p_2}{p_1}\right]^{\frac{1}{n}} = V_p\left[1 + m - m\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}\right]$$

The volumetric efficiency is the ratio $V_1 \div V_p = 1 + m - m\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$ where $n$ for an adiabatic process with ammonia is about 1.3, but for usual expansion and compression it is better to take the value as 1.28.

the actual weight of the gas pumped multiplied by the specific volume of the saturated gas at the suction pressure by the piston displacement. The real volumetric efficiency takes into account the superheating of the suction gas due to the effects of the cylinder walls, piston, valves, etc., which are always at some temperature intermediate between the temperature of the discharge and the suction gas. This effect of the cylinder walls cannot be shown on the indicator diagram. Its value for ammonia and carbon dioxide is shown in Figs. 22 and 23. The superheating of the suction gas is an absolute loss, both as regards the work done on the gas and the capacity of the compressor. The uniflow principle of compressor design, and the use of stage compression are the best means of reducing this loss.

**Speed of the Compressor.**—For considerable time the piston speed has been limited to about 400 ft. per minute, but this limitation has not been general, since, for example, in air compression a speed of 600 ft. per minute has been used with success, and steam and gas engine practice has permitted 800 ft. per minute. The limit of the piston speed had been nothing more than the ability to lubricate properly both the piston and the cylinder walls.

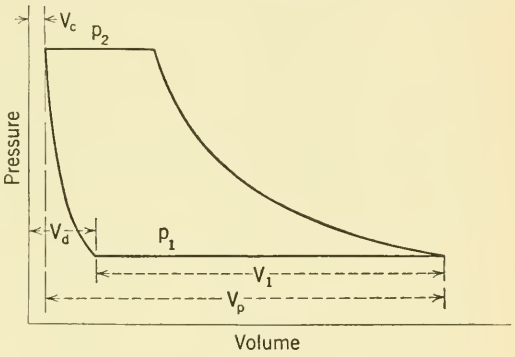



Fig. 21.—Indicator Card.

The rotative speed, however, is another matter, as the number of strokes determines the number of times the valves have to open and close per minute. In order to increase the rotative speed the compressor needs to have a light weight valve and one that has a liberal net opening for the passage of the gas. As the number of revolutions per minute increases, the amount of the valve lift has to diminish, or pounding of the valve will be the result. The outcome of these factors has been the almost universal use of the plate, the ring plate or the ribbon valve. With the exception of the sleeve valve compressor (Fig. 47), mechanically operated valves are never used in refrigeration, but two types of "inertia" valves are used: the balanced piston valve used in the vertical single-acting compressor, and the special suction valve of the ring-plate type used by the Arctic Ice Machine Company in their horizontal double-single-acting compressors.

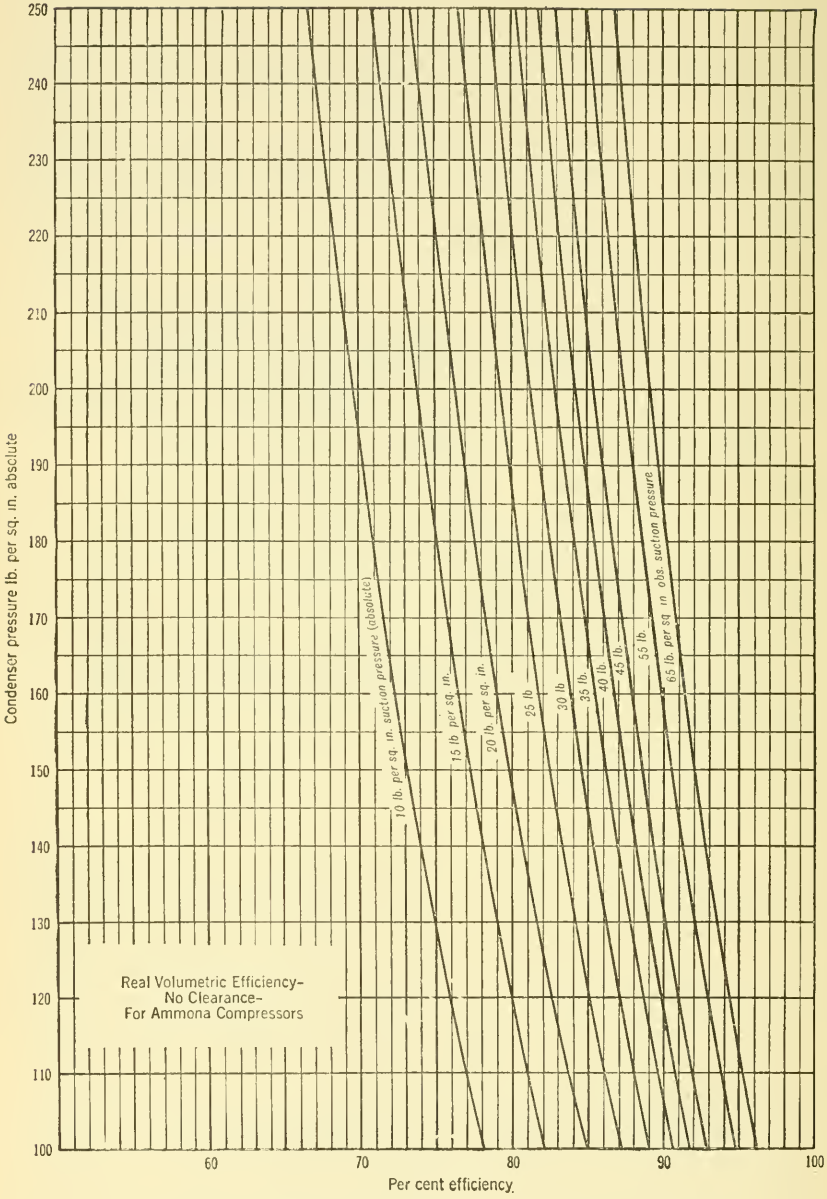



FIG. 22.—Volumetric Efficiency of Ammonia Compressors.

**The Water Jacket.**—The effect of the water jacket for ammonia compressors has been given considerable attention, although until 1921 it was a matter of controversy between manufacturers and theorists.

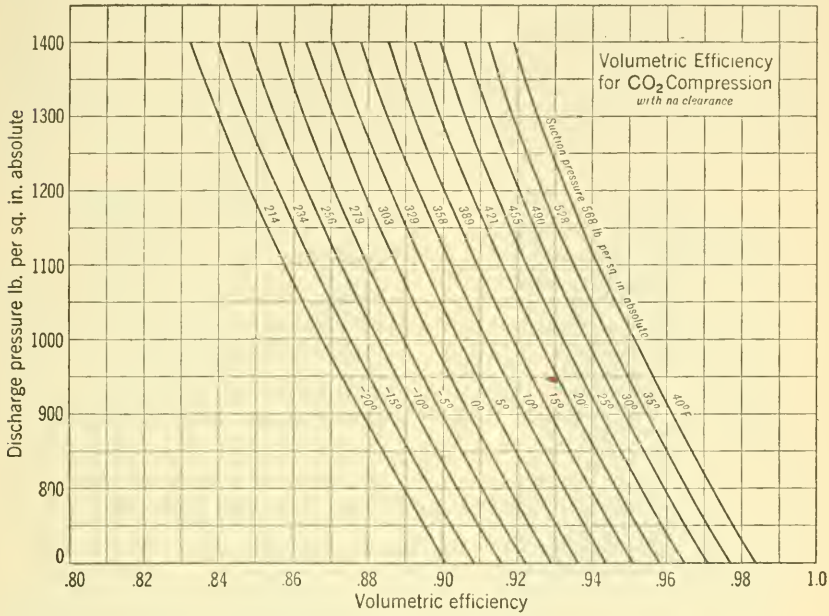



Fig. 23.—Volumetric Efficiency of Carbonic Compressors.

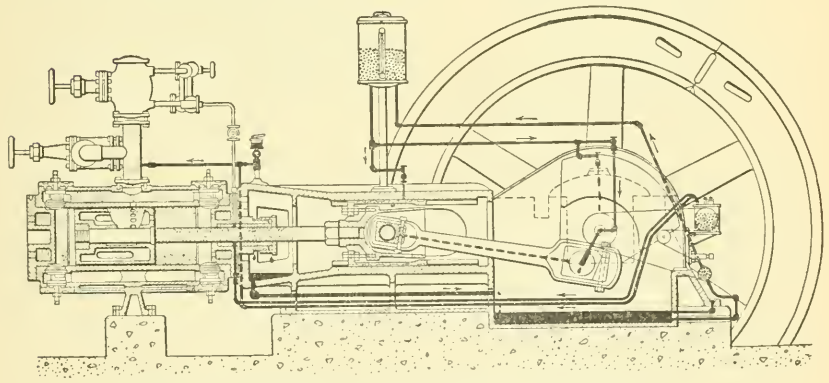


Fig. 24.—Lubrication; the Horizontal Compressor.

The accepted opinion has been that the water jacket is of little value as regards its effect on the compression line, but that its advantage is mainly in reducing the discharge gas temperature and in preventing the overheating of the compressor cylinder.

Because of the fact that the suction gas enters at a temperature much lower than that of the jacket water, the water jacket is designed slightly different from that in the air compressor. The vertical single-acting ammonia compressor is water jacketed the last third of the stroke, whereas the horizontal double-acting compressor is jacketed frequently only at the heads and the ports. The carbonic compressor usually has not been jacketed at all because of the comparatively small amount of rise of temperature; and the majority of the fractional tonnage compressors, using sulphur dioxide or methyl chloride, are designed for air cooling.

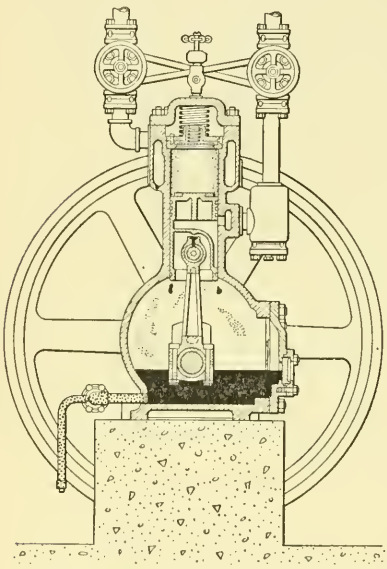


Fig. 25.—Lubrication; the Vertical Enclosed Compressor of 25 Tons and Less.

The most complete tests on the effect of water jackets are those of Walter Fisher.[3] These tests show the relative effect on the compressor with and without the water jacket. For example it is shown that the real volumetric efficiency is increased from about 1.5 to 4.0 per cent by having the water jacket.

**Lubrication.**—With the changes in design have come radical changes in the manner of lubrication. With the horizontal and the semi-enclosed vertical compressor the design has developed into an automatic forced feed system for each surface and bearing, embodying also the oil filter system (except for cylinder lubrication) in order to keep the oil in good condition. As the speed of the compressor increases, the need of a positive, correct method of lubrication is very evident. In the case of the crosshead pin the lubrication is frequently obtained by running the oil pipe along the connecting rod (Fig. 24) in some suitable manner from the crankshaft, or by drilling along the entire length of the rod as shown in Fig. 18.

The enclosed type of single-acting vertical compressor has been lubricated only by the splash method (Fig. 25) and this means is still used in the smaller compressors. In the splash system of lubrication the oil in the crankcase is splashed at each revolution of the crank.

[3] Walther Fisher—Der Einfluss des Kuhlwassermantels an Kompressions-Kältemaschinen—Verein deutsche Ingeniere, 1921.

The splash method of lubrication is excellent for certain conditions, and it is the cheapest of all methods used for continuous work. There are two objections to its use. The first objection is that at the higher speeds the lubrication may not be positive enough because cavitation results in an inefficient formation of the splash with the result that lubrication may be poor or even a failure. The second objection is that lubrication depends on the continuous presence of oil in the crankcase. Certain operating conditions assist in the loss of this oil through

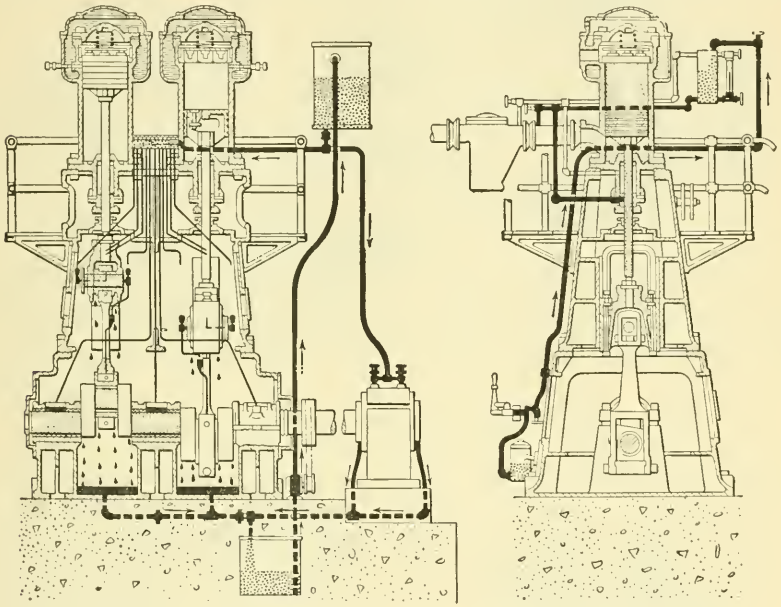


Fig. 26.—Lubrication; the Vertical, Open Frame, Compressor.

the piston into the oil separator, leaving the crankcase with insufficient oil. There is also the possibility of pumping oil during normal operation, but this danger is lessened by care in the design of the rings in the piston.

The result of this lack of positive lubrication has resulted in the design of the forced feed type of pump lubricator, use being made of the oil filter and (usually) a spy glass in the pipe line to the several bearings. Figure 15 shows this type of lubrication design as applied to the smaller capacity of compressor, say from 2 to 12 tons of refrigeration and designed for speeds varying from 530 r.p.m. at 2 tons to 400 r.p.m. at 12 tons. This design lacks the filter and the sight feed and is entirely within the crankcase, but the action may be made positive,

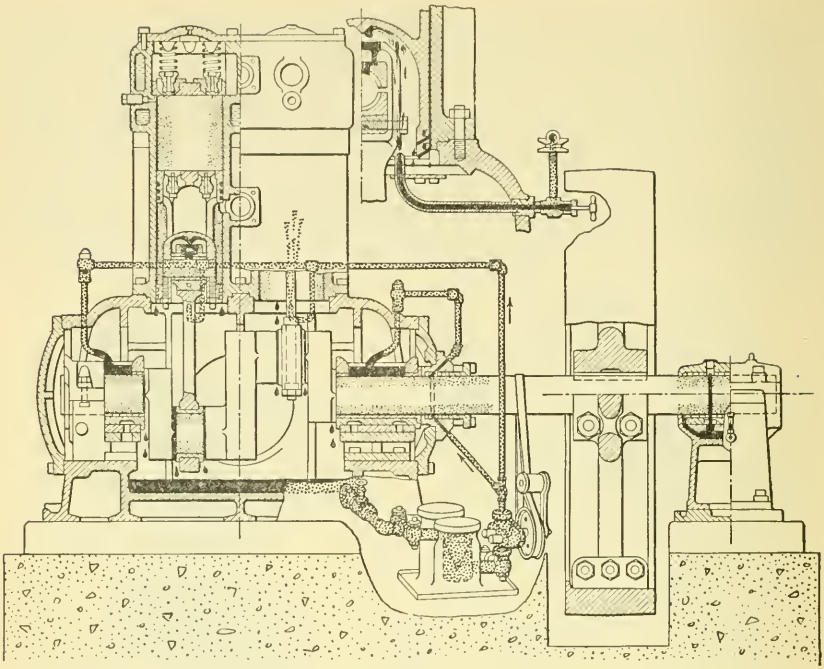


Fig. 27.—Lubrication; the Large, Enclosed-type, Ammonia Compressor.

except for clogging, by the use of some form of rotary pump and by having no suction lift to the pump itself. Figure 27 shows a design typical of the more elaborate method of lubrication, such as would be found in the larger compressors, one which has a nozzle device for spraying the cylinder, provided with a nozzle clearer, to be used at regular intervals to prevent the clogging of the nozzle, and a separate pipe line for the oil to every important bearing. Figure 26 shows the method used in the vertical open-frame compressor.

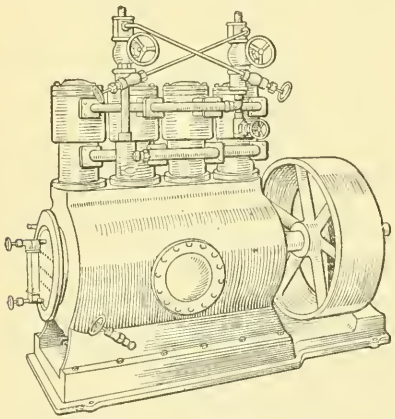


Fig. 28.—The Eccentric Driven Compressor.

**Number of Cylinders.**—The use of multiple cylinders has been restricted in the United States to two in number as a rule, although the

three-cylinder compressor has been built to a limited extent. This is apparently a contradiction of the practice in Europe and particularly in Great Britain, where they are pushing the type using the three- and the four-cylinder design of compressor especially in the larger sizes, as shown by Fig. 18. In Great Britain the tendency appears to be for increase of the capacity of the compressor by adding cylinders, but in the enclosed design only and not the open frame semi-enclosed type of single-acting vertical compressor—the latter being distinctively an American design. There is another advantage in the use of multiple cylinders from the viewpoint of balance. With one or two cylinders no amount of counterbalance will entirely balance the effect of reciprocation whereas the three- or four-cylinder machine (Figs. 18, 19, 28) may be entirely balanced except for a moment which will need to be carried by the bed plate or the foundation. It would seem then that if the compressor speed is to be increased decidedly, there will be an advantage in the use of three cylinders on the same crankshaft.

**Theoretical Displacement of the Compressor.**—The theoretical displacement of the compressor may be found in the following manner:

The refrigerating effect of the volatile liquid per 1.0 lb. of refrigerant is $(i''_3 - i'_1)$ (Fig. 1). The piston displacement per ton of refrigeration per minute

$$\text{P.D.} = \frac{200}{i''_3 - i'_1} \times V''_3,$$

where $V_3''$ is the specific volume of saturated ammonia (or refrigerant) at the pressure *3*. If the real volumetric efficiency of the compressor (expressed as a decimal) is $E_s$, then the total required piston displacement becomes

$$\text{P.D.} = \frac{200}{i''_3 - i'_1} \times V''_3 \div E_s$$

assuming tight valves, rings and no clearance. If $E_v$ is the volumetric efficiency due to *clearance only*, then the piston displacement required with clearance is,

$$\text{P.D.} = \frac{200}{i''_3 - i'_1} \times \frac{V''_3}{E_v \times E_s}$$

There is no reliable method of calculating the real volumetric efficiency although the formula $E_s = 1 - \frac{t_4 - t_3}{1330}$ where $(t_4 - t_3)$ is the rise of temperature during the compression stroke has been proposed

as an empirical formula based on tests conducted by the York Manufacturing Co. (1903) and gives approximate results. The temperature at the point 4 can be found from the thermodynamic equation for exponential processes (Chapter I):

$$T_b = T_a\left(\frac{p_b}{p_a}\right)^{\frac{n-1}{n}}$$

where the temperatures are absolute and $t_b = T_b - 460$.

Figures 29 and 30 have been calculated, using the equations for volumetric efficiency and piston displacement.

**Horse Power per Ton of Refrigeration.**—The horse power per ton of refrigeration is a very important quantity, and the theoretical value may be found in the following manner:

The work during any exponential compression cycle is:

$$W = \frac{n}{n-1}p_1v_1\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]$$

where $n$ is taken as approximately 1.28 for ammonia, and for carbon dioxide, and $p_1$ and $p_2$ are the absolute pressures at the beginning and the end of compression in pounds per square foot and $v_1$ is the volume in cubic feet at the beginning of compression. If $v_1$ is taken in the formula as the volume of the piston displacement per ton of refrigeration per minute, and the result is divided by 33,000, the final result is the horse power per ton of refrigeration. For ammonia and carbon dioxide, and several other refrigerants, there is also the $P$–$I$ diagram (Fig. 164, etc.) from which it is possible to get the work of compression per one pound of the refrigerant, by reading the value of the thermal potential $I$ at the two points corresponding to the beginning and the end of compression. The theoretical indicated horse power (i.hp.) can then be found by multiplying by a suitable constant and by the number of pounds of refrigerant required per ton of refrigeration per minute. The formula becomes

$$\text{i.hp.} = \frac{200}{i''_3 - i'_1} \times \frac{777.6}{33{,}000} \times (i_4 - i_3) = 4.713\left(\frac{i_4 - i_3}{i''_3 - i'_1}\right)$$

where $i_3'' - i_1'$ is the net refrigeration per 1.0 lb. of refrigerant.

Figures 31 and 32 give values for the horse power per ton of refrigeration for ammonia and carbon dioxide compressors.

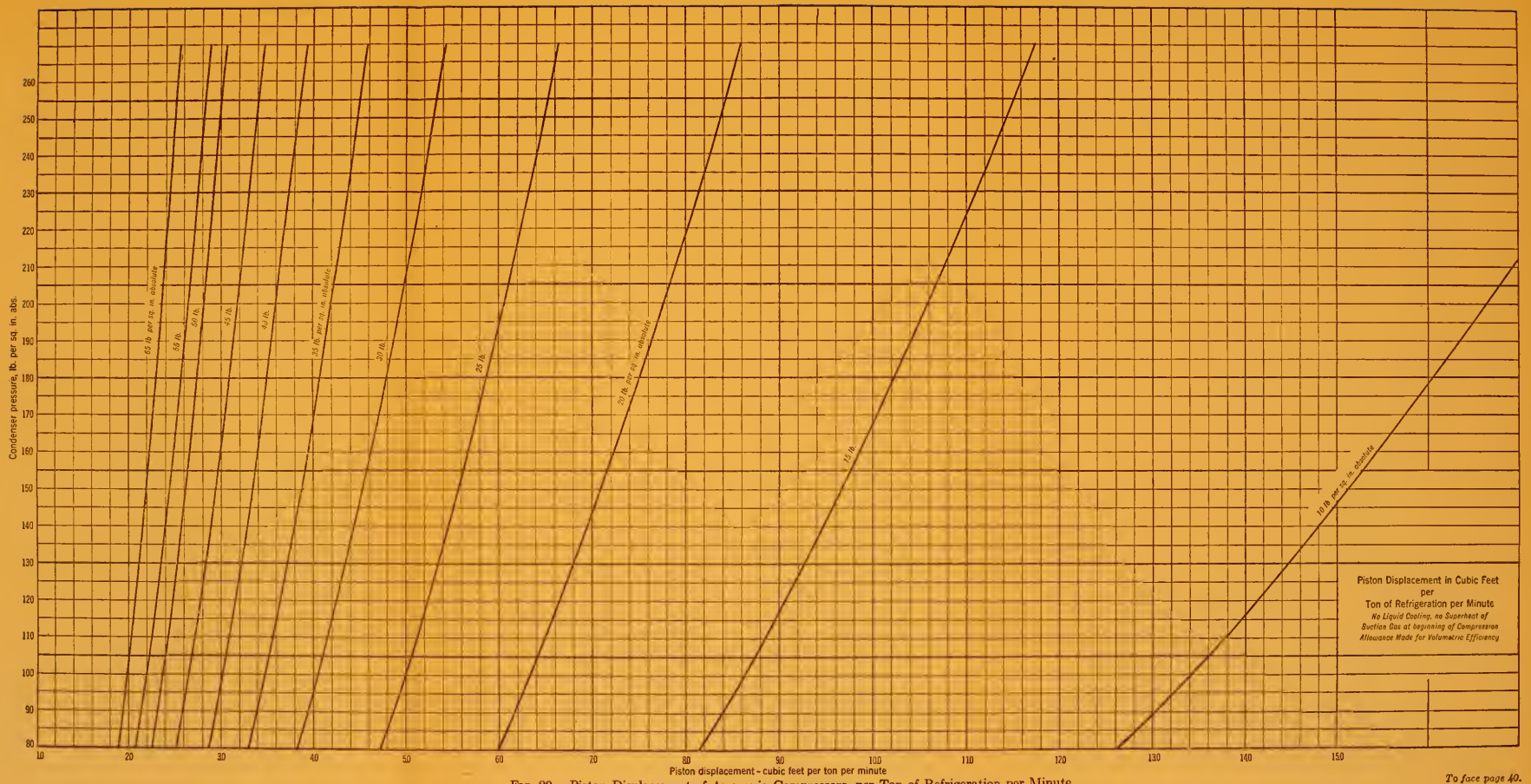



FIG. 29.—Piston Displacement of Ammonia Compressors, per Ton of Refrigeration per Minute.

*To face page 40.*

## CARBON DIOXIDE COMPRESSORS

The use of carbonic refrigeration is increasing in the United States but not in proportion to the amount of new tonnage sold. Contrary to the conditions in Great Britain, where carbonic refrigeration is required by law under specific conditions, carbonic refrigeration is making gains in the United States practically on its own merits. It is a real *safety* refrigerant, and as such is particularly advantageous for use in theaters, hotels, apartment houses and other places where the

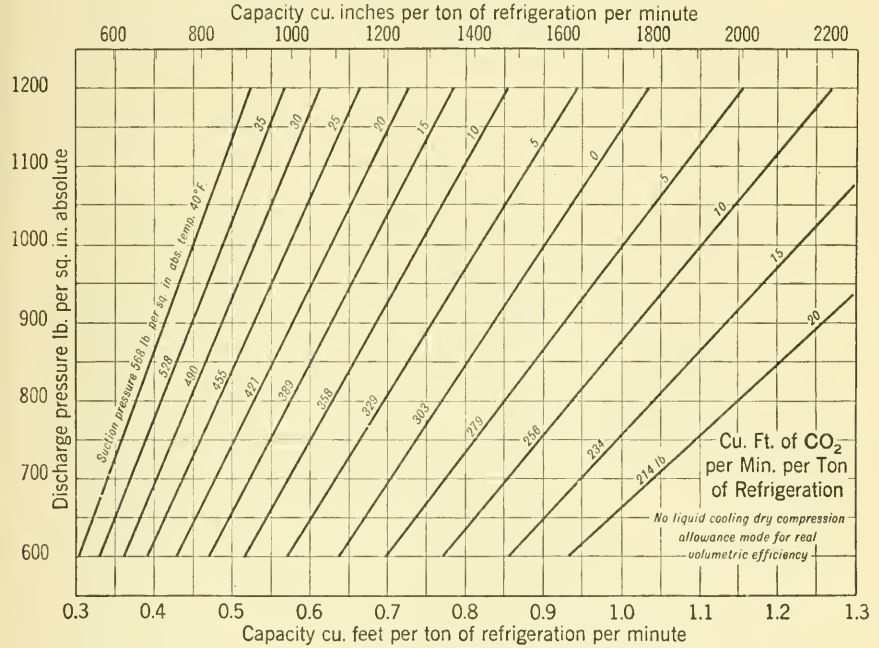



FIG. 30.—Piston Displacement of Carbonic Compressors, per Ton of Refrigeration per Minute.

public is exposed, and the risk due to panics might be a positive danger. A small amount (even a trace) of ammonia in the air is a menace, whereas it has been demonstrated that carbon dioxide in large quantities is not injurious except in that it excludes oxygen, a condition that might finally cause suffocation.

Carbonic refrigeration has suffered because of an improper understanding of the refrigerant. As the critical temperature is 88.0 deg. F. (corresponding to a pressure of 1072 lb. per square inch) it is evident that temperatures of the condensing water of 80 deg. F. and over, mean

that the refrigerant is *not* condensed, but that does not mean that no refrigeration is possible under these conditions. It means only that less useful refrigeration is possible per pound in the same manner that less refrigeration is possible when a hot liquid passes the expansion valve in ammonia compressors than would be the case with a cold liquid. It seldom occurs that the quality ($x$) of the gas after the expansion valve even under these extreme conditions is less than 0.5, as is seen on referring to the $P$–$I$ diagram for $CO_2$ (Fig. 164).

There is another factor antagonistic to the increased use of carbonic machines. The pressures involved are heavy, varying from 1000 to 1200 lb. or more on the high-pressure side and from 300 to 600 lb. per square inch on the low-pressure side. Losses of gas to the atmosphere have been excessive, and leaks past the valves, piston rings, etc., have been very high even with new machines. It appears then that a successful carbon dioxide compressor and system must be properly designed, and the materials entering into its construction and the *workmanship* must be satisfactory. Fortunately autogenous welding has become so common and reliable, especially for small pipe work, that fittings can be eliminated to a large extent, and the gas losses can be reduced accordingly. This is becoming true in the United States for the double-pipe condenser (Fig. 66), the low-pressure piping and to some extent the high-pressure part of the system.

**Design.**—Carbonic compressor cylinders are made of semi-steel (instead of steel forgings as formerly) cast solid and bored out for the cylinder bore, the ports and valve cages. As a rule the cylinder, crank-end cylinder head and the stuffing box are made of a single casting, thereby permitting better alignment, and eliminating one joint and the source of another probable leak. The head cylinder cover is held in place by studs or through bolts. The remainder of the construction is very much like the design for ammonia compressors, as regards construction of the piston rod, connecting rod, crosshead, etc. For example, the piston rods, connecting rods and crankshaft are of forged steel, the piston rod being hardened and ground to size. The crosshead is of semi-steel and the pin is of steel hardened and ground. The brasses are of phosphor-bronze and are made adjustable. Because of the small ratio of the condenser to the suction pressure $\left(\frac{p_2}{p_1}\right)$ the discharge gas is not high in temperature and water jackets for the cylinder are never used, but on account of the very heavy discharge pressure, and the desire to keep the so-called hoop tension in the cylinder as low as practical—the cylinder diameter is kept as small as possible. The necessary piston displacement of the compressor is obtained by making the stroke

long in proportion to the piston diameter, the ratio being from $3\frac{1}{2}$ to 4, or more than twice the ratio usually found in similar ammonia compressors. Except for marine and some small enclosed machines the compressor is of the horizontal, double-acting type, using metallic packing and the same design of lantern as in ammonia compressors. Also, because of the small ratio of pressures, clearance does not affect the capacity as much as it would in the case of ammonia, and the capacity is not reduced very much by wire drawing in the suction because of the usual high suction pressures of 300 to 450 lb. In consequence a drop of 5 lb. due to friction would not be as serious as would an equal drop in the case of ammonia with a suction pressure of 15 lb. gage. The valves may be poppet valves in 45-degree seats, or they may be ring plate valves, in which case, the disc is frequently made of chrome vanadium steel, and the lift is limited to as little as $\frac{1}{16}$ in.

Although the ratio of the condenser to the suction pressure is not great, being but seldom more than 1 : 4, yet the differential pressure is very heavy—being from 900 to 1000 lb. The cylinder and the piston should be ground carefully to a true surface and the cylinder should be lapped out. The piston is supplied with cast iron or semi-steel snap rings which may be 3, 4, 5, or 6 in number. The H. J. West design of vertical enclosed type of compressor which is designed for rotative speeds of 400 to 500 r.p.m. has 7 snap rings in addition to 2 scrapper rings placed on the crosshead part of the trunk piston. The tolerance for cylinder bore is 0.002 in. for diameter and parallel to 0.001 in. per foot of length. The piston diameter is specified to be at least 0.0005 in. per inch of diameter less than the nominal diameter of the piston, but in no case is the difference between the cylinder diameter and the diameter of the piston to be less than 0.0007 in. per inch of diameter, nor more than 0.0015 in. per inch of diameter. It is made round and parallel within the limits of 0.001 in. The piston rod is round within the limits of 0.002 in. and parallel within the limits of 0.003 in. The total clearance between the piston head and the ends of the cylinder is about $\frac{1}{8}$ in. The entire system is designed to be tight under a pressure of 3000 lb. per square inch of water and 1350 lb. air pressure under water.

The stuffiing box is still designed to use leather cups and other soft packing combined with brass boxes and washers or other forms of distance pieces, but this design is giving way to metallic packing. When the soft packing is used, oil is fed into the lantern at a pressure greater than the gas pressure in the gland and the stuffing box is well supplied with oil so that the surplus works along the rod and into the cylinder, lubricating the piston in the cylinder. Metallic packing is usually freer from leaks provided the rod is turned true and is centrally located

in the machine. There is less danger of burning the packing than in the case of the leather cups.

The carbonic compressor has two factors influencing its capacity, one of which, the leakage factor, has been discussed already. Leaks past the valves and the piston are reduced to a minimum by the use of good material and design and very careful machine work. The second factor is the large value of the thermal potential ($t$) of the liquid and the relatively low value of the latent heat of vaporization. Referring again to the $P$–$I$ diagram (Fig. 164), it will be seen that the value of "$x$" is quite large even for a nominal range between the suction and the condenser pressures. The result is that whereas the power requirement would be the same for all refrigerants if the liquid is cooled down first to the temperature of the evaporator, yet, as a matter of fact, there are large variations. Tests reported by Professor Carl Linde[4] showed, with 72 deg. F. condenser temperature and 15 degrees brine, that the carbonic machine was only equal to 82 per cent of the coefficient of performance of an ammonia machine operating under similar conditions, and if the condenser temperature was increased to 95 degrees the coefficient of performance dropped off to 50 per cent. Willcox and Hodgdon showed[5] that the brake horse power per ton of refrigeration (i.hp. plus ten per cent) with − 10 deg. F. evaporating temperature of the liquid carbon dioxide was 1.67 with 55 deg. F. liquid $CO_2$ at the expansion valve, 2.0 with 65 degrees, 2.39 with 75 degrees, 2.85 with 85 degrees and 3.44 b.hp. with 95 degree "liquid" at the expansion valve per ton of refrigeration. These values are considerably higher than would be required with ammonia. The natural consequence is that ammonia is used to the exclusion of carbon dioxide in all large cold-storage warehouses or in ice plants where the unit power costs—the cost of power per ton of refrigeration per 24 hours—or the power required per ton of ice delivered on the platform—is an important factor. On the other hand, some hotels make their own ice, with the refrigerating plant installed for cooling drinking water, and for kitchen and storage boxes. Therefore carbonic compressors are frequently used for this kind of installation, even though the cost of making the ice is greater than it would be with ammonia as the refrigerant. Figs. 23, 30 and 32 give an idea of the volumetric efficiency, the cubic feet of piston displacement per minute per ton, and the brake horse power per ton of refrigeration. Fear of ammonia is so great with certain individuals that carbonic compressors are becoming more popular in America and without doubt improved design will make them even more so. Installa-

[4] Zeitschrift für die Gesammte Kälte-Industrie, Jan. 28, 1895.

[5] Willcox and Hodgdon, Cold Storage and Ice Association, Jan. 27, 1914.

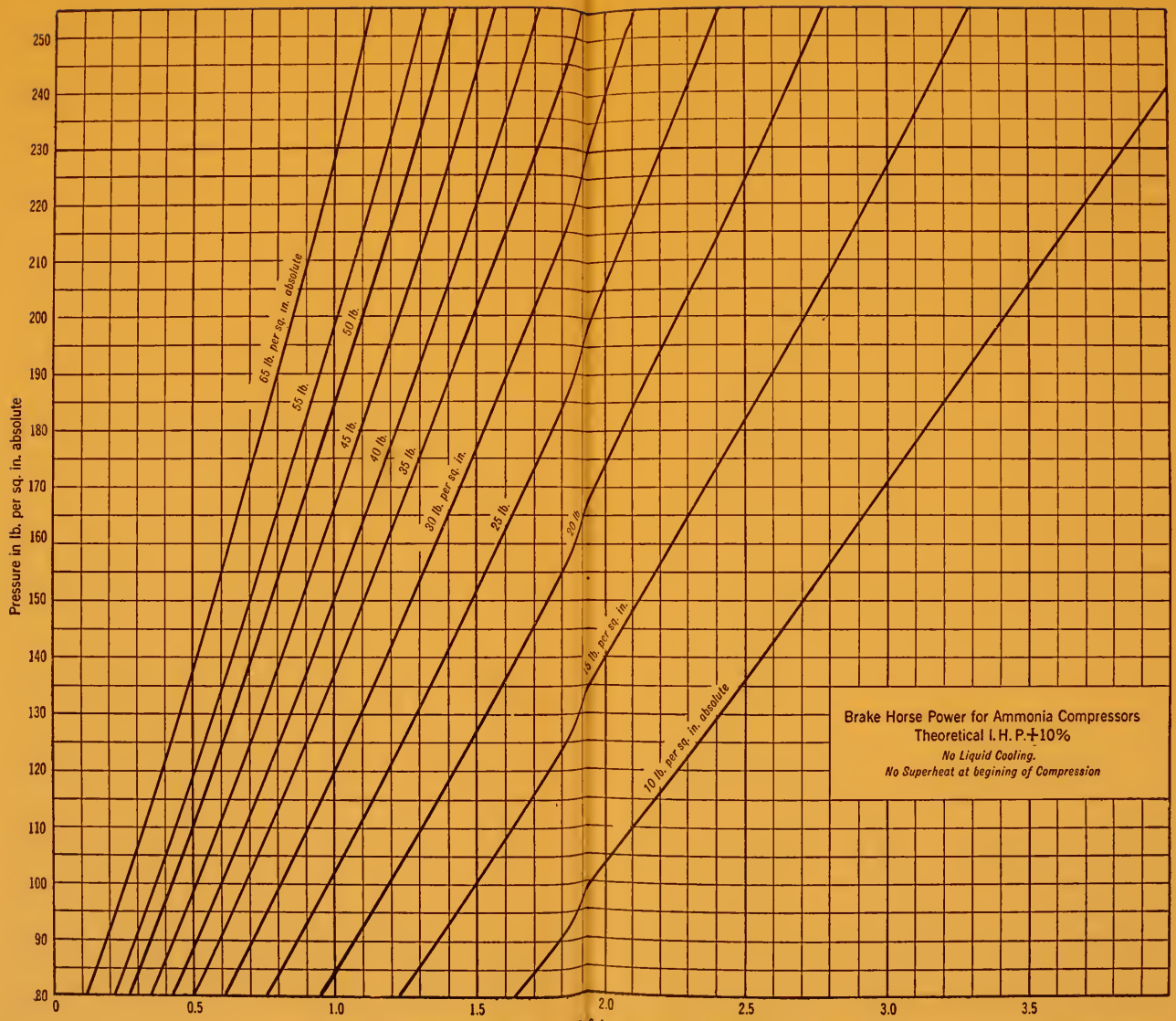



FIG. 31.—Horsepower per Ton of Refrigeration for Ammonia Compressors.

*To face page 44.*

tions are being made even in Florida and the Gulf States, though the cooling water necessary for this type of machine is of a temperature that would seem prohibitive in such latitudes.

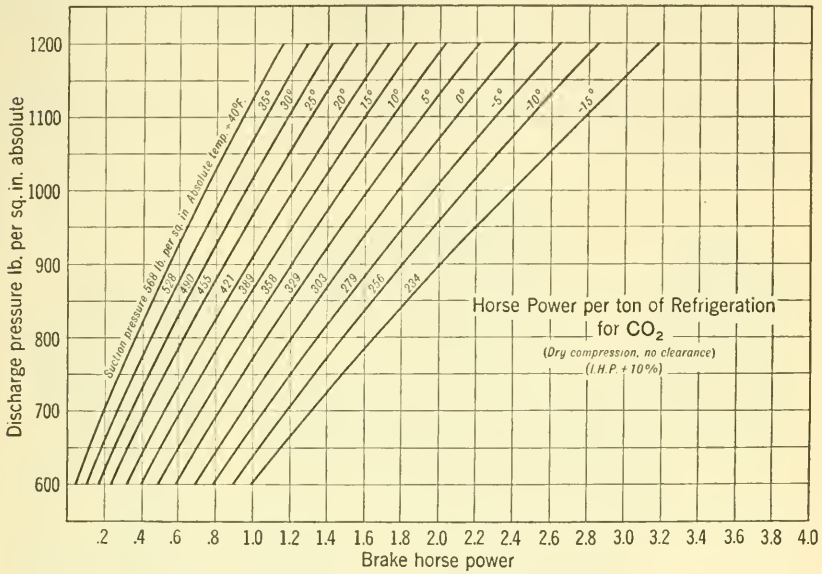



Fig. 32.—Horse power per Ton of Refrigeration for Carbonic Compressors.

The carbonic compressor has been used for many years in the merchant marine to the exclusion of all others, on account of the Underwriters' rules. Lately, however, there has been a tendency for the use

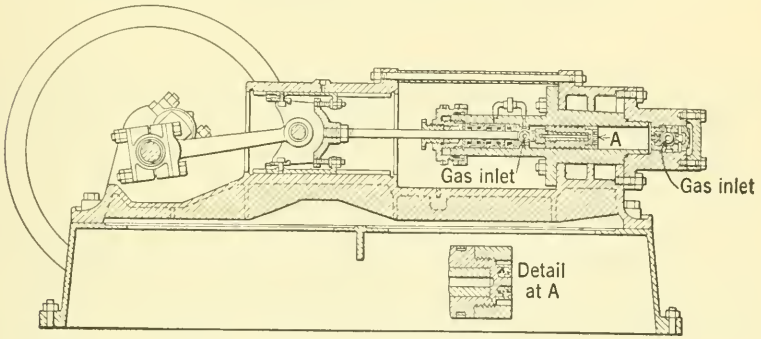



Fig. 33.—The Carbonic Compressor.

of ammonia compressors located outside the main engine room and preferably convenient to the deck, especially if the size of the installation warrants the employment of a refrigerating engineer. In the smaller

sizes, where the regular watch engineer has to operate the refrigerating machine also, the installation has to be in the main engine room. Where ammonia compressors are used on shipboard wrought iron and steel condensers have to be selected, in which case corrosion is more rapid than would be the case of copper condensers which are employed so extensively for carbonic installations.

The carbon dioxide compressor can produce low temperatures with considerable ease and in Great Britain they have been built in two stages for that purpose. According to G. W. Daniels,[6] a battery of 4

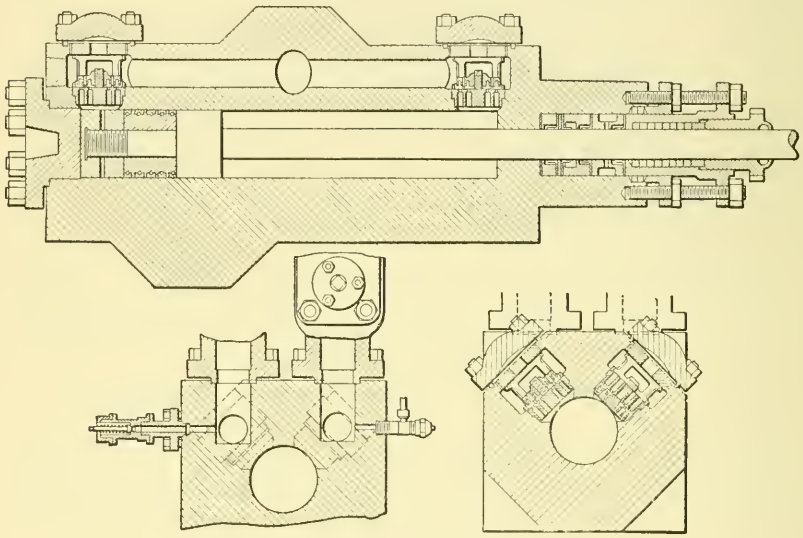


Fig. 34.—The Carbonic Compressor.

2-stage carbonic compressors operating at − 58 deg. F. has been very successful. Figures 33 to 35 inclusive show representative carbonic compressors and Fig. 36 gives results of tests on a small marine type carbonic compressor.

**The Sulphur Dioxide Compressor.**—The compressor using sulphur dioxide as a refrigerant has not been used to any extent in the United States except in the case of household machines, although numerous attempts have been made to establish its use. This fact is somewhat surprising when one recollects that in Germany alone there are 17 firms listed as building $SO_2$ machines, including A. Borsig, Quire and Co., Gesellschaft für Lindes Eismaschinen, Maschinenbau-Anstealt Humbolt, etc., built in capacities up to 320,000 cal. per hour or about 106 tons of

[6] G. W. Daniels, Refrigeration in the Chemical Industry. A. J. Rayment, 1925.

refrigeration, as compared with 27 German firms building ammonia and 29 building carbonic compressors. In a letter in the May, 1925, issue of Cold Storage, W. S. Douglas of Wm. Douglas & Sons says that 90 per cent of all $SO_2$ refrigerators operate at a brine temperature of 15 to 25 deg. F. He says the use of $SO_2$ as a refrigerant is increasing steadily and, in his opinion, this is the best refrigerant for small machines on account of the low pressure, the lubricating properties and the ease in handling.

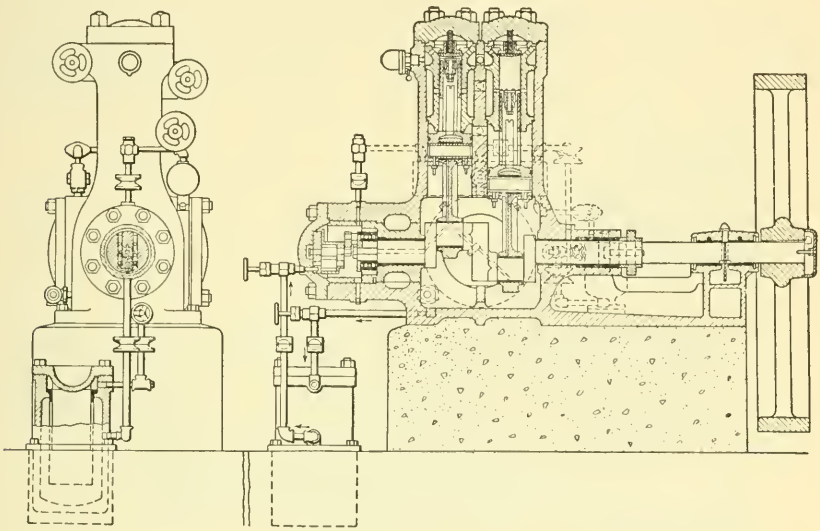


FIG. 35.—The Enclosed Type Carbonic Compressor.

The sulphur dioxide compressor requires some 2.5 times the piston displacement of the ammonia compressor, and the design of the German compressors very closely resembles that for ammonia. Because of the larger volume of the gas handled, and the relatively low unit pressures the valve is likely to be of the flap (Gutermuth) type as used in the larger machines of the A. Borsig design or of the modified plate-valve type. As sulphur dioxide is corrosive when any moisture is present the tendency is to restrict the amount of evaporator surface as much as possible, and in consequence the direct expansion system is not popular but a self-contained brine system is more likely to be used. In the United States it seems now very unlikely that sulphur dioxide will be used to any extent except in small machines of the household or the $\frac{1}{6}$-ton or the so-called ice cream cabinet size or in the case of the *A–S* (dumb-bell) (Fig. 37 and Table 2) compressor. In this compressor the casing is

hermetically sealed and cannot be opened for repairs except at the factory. Also, except in the small sized compressor, sulphur dioxide has practically nothing in its favor, and it is more than likely that it will be abandoned in favor of methyl chloride, butane or some other low-pressure refrigerant. All of these have the advantages, except for self-lubrication, and none of the disadvantages of sulphur dioxide, although it is true that sulphur dioxide ($SO_2$) leaks, being easily detected, can be easily remedied.

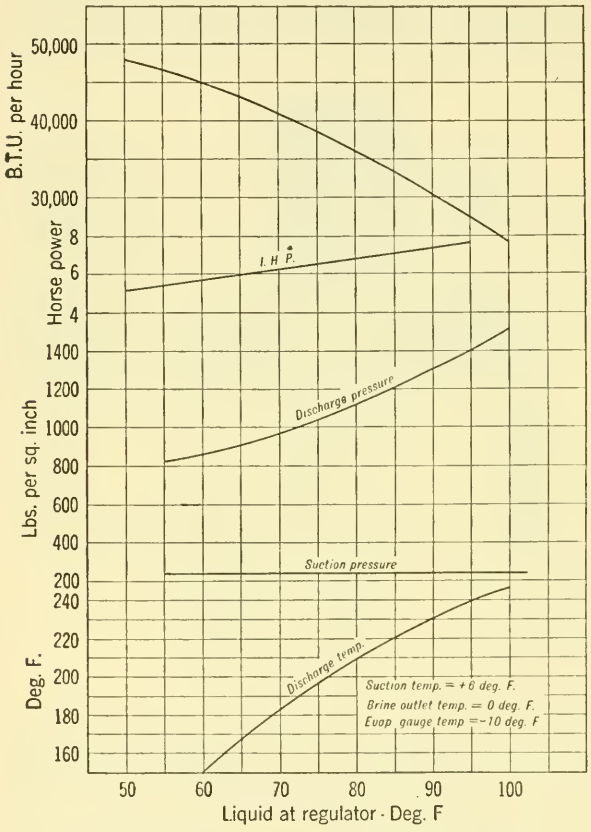



Fig. 36.—Test Performance of Carbonic Compressors.

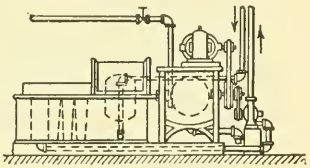
**The Rotary Compressor.** — The rotary compressor is the ultimate goal of many engineers as it will permit high rotative speeds and therefore direct connection to a cheap type of electric motor. The piston displacement per ton of refrigeration need not cause any concern as the displacement is easily cared for in such a design, and the refrigerant may be chosen that is best suited to rotary compression without regard to other factors.

**The Blade Type.**—The most common type of rotary is the blade type to operate eccentric in the casing and so arranged as to depend on centrifugal force to keep the blade always in contact with the casing. Such a machine using ammonia has been described by Roloff,[7] but no such compressor has been successful. Other compressors (Fig. 38), using ethyl chloride, have been successful on test. Results of these

[7] Roloff, Transactions of the A. S. M. E., 1918.

tests reported by Herter[8] indicate that the economy of operation is not so great as for ammonia, although it must be remembered that the small machine is seldom as efficient as the larger one. His figures are as follows:

| | | | | |
|---|---|---|---|---|
| Refrigerant evaporation temperature, degrees F. | 30 | 19.9 | 2.7 | −9.6 |
| 20-ton ammonia compressor (horse power per ton) | 1.1 | 1.3 | 1.6 | 2.0 |
| 2-ton Clothel compressor (horse power per ton) | 2.3 | 2.8 | 3.3 | 4.4 |
| $\frac{1}{10}$-ton Williams compressor (horse power per ton) | 4.3 | 5.8 | 11.0 | 22.0 |



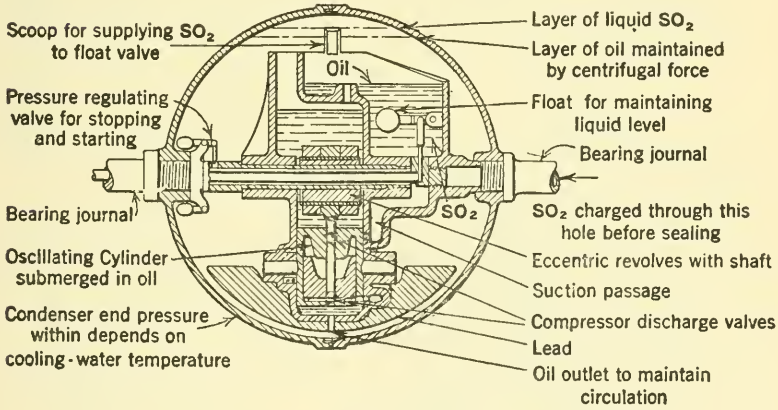

Scoop for supplying $SO_2$ to float valve
Oil
Layer of liquid $SO_2$
Layer of oil maintained by centrifugal force
Pressure regulating valve for stopping and starting
Float for maintaining liquid level
Bearing journal
Bearing journal
$SO_2$
$SO_2$ charged through this hole before sealing
Oscillating Cylinder submerged in oil
Eccentric revolves with shaft
Suction passage
Condenser end pressure within depends on cooling-water temperature
Compressor discharge valves
Lead
Oil outlet to maintain circulation

Fig. 37.—Audiffern-Singrun Sulphur Dioxide Compressor.

**The Centrifugal Compressor.**—If the centrifugal compressor is used, the design principle is that of developing a velocity to the gas, and then by slowing up this velocity of converting velocity head into pressure head. Such a method has been used to compress air by the use of a very large number of stages. For refrigeration the compression of carbon dioxide and ammonia is practically impossible, but by the judicious selection of a heavy gas of low condensing pressure the same end can be reached. This has been accomplished by the Carrier

[8] C. H. Herter, Refrigerating World, Sept., 1922.

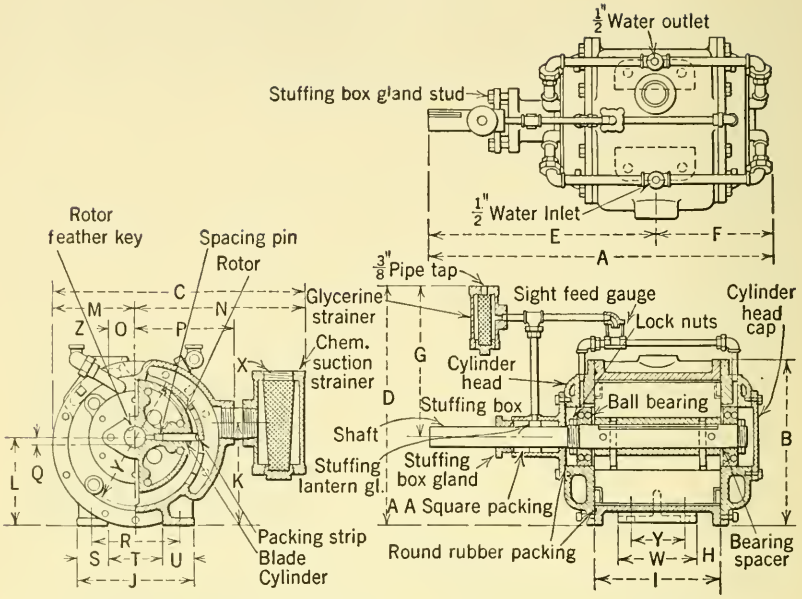



Fig. 38.—The Clothel Rotary Compressor.

| Size | Dimensions in Inches | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| Tons | A | B | C | D | E | F | G | H |
| ½ | 15½ | 6⅝ | 12 5/16 | 14¾ | 10 9/16 | 4 15/16 | 11 | 13/32 |
| 1 | 25⅛ | 10⅝ | 17 5/16 | 18 | 16¼ | 8¾ | 12 | 9/16 |
| 2 | 32¾ | 16 | 24⅜ | 23 | 21⅝ | 11⅛ | 14½ | 1 1/16 |
| Tons | I | J | K | L | M | N | O | P |
| ½ | 3½ | 6 | 3 13/32 | 3¾ | 3 5/16 | 9 | 1½ | 4 |
| 1 | 9 | 10 | 6⅞ | 6 | 5 5/16 | 12 | 1¾ | 6¼ |
| 2 | 12 | 11½ | 9⅞ | 8½ | 8 | 16⅜ | 2½ | 9½ |
| Tons | Q | R | S | T | U | V | W | X |
| ½ | 7/32 | 4¾ | 1½ | 3 | 1½ | 1¼ | 2½ | 1 |
| 1 | 7/16 | 8 | 2¼ | 5½ | 2¼ | 4¼ | 5¾ | 2 |
| 2 | 21/32 | 9¼ | 3 | 5½ | 3 | 5¾ | 7½ | 2½ |
| Tons | Y | Z | A–A | R. P. M. | Net Weight, Pounds | | | |
| ½ | 5 | ¾ | 5/16 | 900 | 65 lbs. | | | |
| 1 | 8 | 1¼ | ⅜ | 350 | 225 lbs. | | | |
| 2 | 12¼ | 2 | ½ | 226 | 580 lbs. | | | |

Engineering Company using as a refrigerant dichlorethylene ($C_2H_2Cl_2$) which condenses at a temperature of 80 deg. F. at a pressure of less than one atmosphere. The compressor (Fig. 39) looks very much like

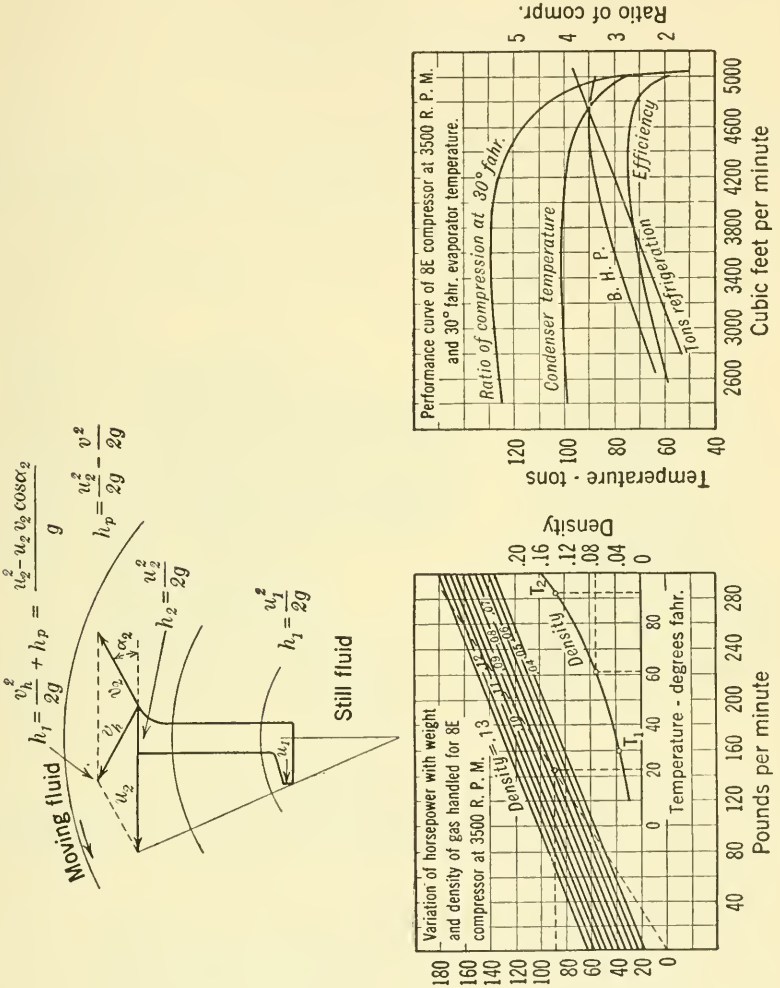

Performance curve of 8E compressor at 3500 R. P. M. and 30° fahr. evaporator temperature.
Ratio of compression at 30° fahr.
Condenser temperature
B. H. P.
Tons refrigeration
Efficiency
Ratio of compr.
Cubic feet per minute
Temperature - tons
Variation of horsepower with weight and density of gas handled for 8E compressor at 3500 R. P. M.
Density
Temperature - degrees fahr.
Pounds per minute
Moving fluid
Still fluid

Fig. 39.

an ordinary water centrifugal pump and is designed for five, six or more stages and about 3600 to 4000 r.p.m. in some sizes. This type of compressor appears to be best suited to the larger sizes, say 75 tons and larger, but it may be possible to build economically in capacities as low as 20 tons. Also the machine is best suited to the higher evaporating

temperatures such as would be found in the cooling of air for theaters and public buildings.[9]

Referring to Fig. 39 it will be seen that the total head produced, the sum of the velocity and static heads, is

$$h_1 = \frac{v_h^2}{2g} + h_p = \frac{u_2^2 - u_2 v_2 \cos a_2}{g}$$

Only a part of the velocity head is available for the production of static pressure, and the usual method of change from velocity to static heads is by means of diffuser blades of gradually increasing cross-sectional area. The efficiency of the conversion is approximately 70 to 80 per cent.

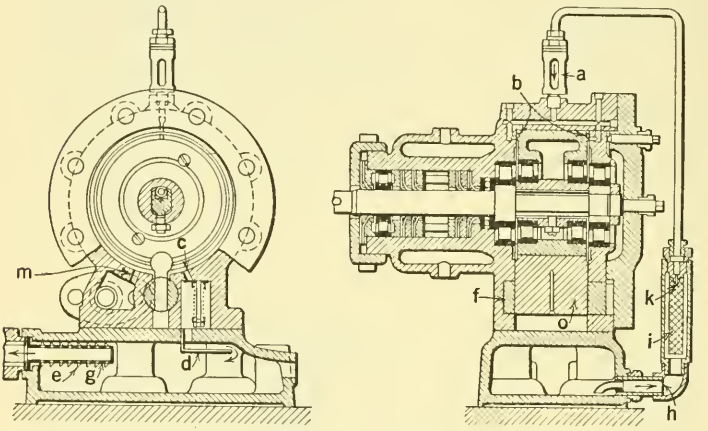



Fig. 40.—The Güttner Pendulum Compressor.

The curves in Fig. 39—taken from an article by Willis Carrier (Refrigerating Engineering, Feb., 1926)—give some idea of the displacement, tonnage and efficiency of a centrifugal compressor using "dieline" as a refrigerant.

**Turbo-Compressors.**—In Germany[9a] the turbo-compressor seems to be successful in the larger sizes. According to Heinz Voight ammonia turbo-compressors are only applicable in capacities greater than 300 tons. The number of rotors for different refrigerants with peripheral speeds of 655 ft. per second are stated to be as follows:

[9] Because the evaporator pressure is so low, a separate means of continually expelling the air leakage must be provided. The float type of expansion valve is used. The compressor, condenser and brine cooler are made self-contained.

[9a] Heinz Voight, Zeitschrift des Vereins Deutscher Ingeniere, Aug. 13, 1927.

Number of Rotors Required for Different Mediums at Peripheral Speeds of 655 Ft. per Second

| Temperature of Vaporization, Deg. F. | $CO_2$ | $NH_3$ | $CH_3Cl$ | $SO_2$ | $C_4H_{10}$ | $C_2H_5Cl$ | $H_2O$ |
|---|---|---|---|---|---|---|---|
| 32 | 2 | 6 | 3 | 3 | 3 | 3 | 17 |
| 14 | 3 | 11 | 4 | 4 | 4 | 4 | |
| − 4 | 4 | 15 | 5 | 5 | 5 | 5 | |
| −22 | 5 | 20 | 6 | 6 | 6 | 6 | |

The guaranteed power consumption of a 2000-ton turbo-compressor for operation at 5 deg. F. evaporation and 86 deg. F. liquefaction temperature at a speed of 6000 r.p.m. is given as 2365 b.hp.

**The Pendulum Type.**—Still another type of rotary, for the use of ammonia, is the Güttner compressor (Fig. 40). This rotary, like the Carrier centrifugal design, does not have contact between the metals of the rotor and the casing, but depends on a film of oil to maintain tightness in the clearance of about 0.0015 in. It is built to operate at 800 to 1500 r.p.m. in 1.0-ton sizes, but it does not depend on the speed for the compression as the action is positive. There is no regular suction valve although a suction check valve is used, but a plate disc is employed as a discharge valve. Tests[10] indicate that the volumetric efficiency is high, but that

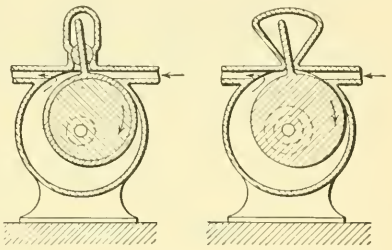


Fig. 41.—Older Designs of Pendulum Compressors.

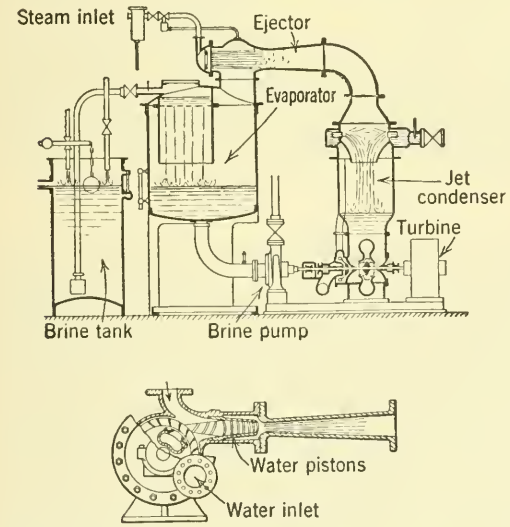



Fig. 42.—The Westinghouse-Leblanc Refrigerating Machine.

[10] H. J. Macintire, A. S. R. E. Journal, March, 1925, Plank, Krause, & Tomm, V. D. I., 1925.

the horse power per ton, as in many small machines, is also high. The older designs operating on this principle are shown in Fig. 41.

**The Water Vapor Refrigerating Machine.**—The water vapor refrigerating machine (the Westinghouse-Leblanc) has been proposed but has not been developed to any extent in the United States. It is a "safety" refrigerating system but it needs a high vacuum and is limited to rather high refrigerating temperatures, such as in the case of air cooling down to 50 or 40 deg. F. The high vacuums are best obtained by means of "water pistons" (Fig. 42) combined with steam ejector nozzles.

In the ejector type of water vapor refrigerating machine the underlying principle is that of a mass in motion communicating motion to another mass. The resulting velocity of the mixture will be expressed by the relation:

$$\frac{mv^2}{MV^2} = \frac{mM}{(m+M)^2}$$

If

$$\frac{M}{m} = y,$$

then

| $y =$ | 1 | 2 | 3 | 4 | 5 |
|---|---|---|---|---|---|
| $\frac{mM}{(m+M)^2} =$ | 0.25 | 0.222 | 0.187 | 0.160 | 0.139 |

where $M$ and $m$ are the two masses, $V$ = the velocity of 1st mass and $v$ = the velocity of 2nd mass. Calculation shows that the amount of ejector steam required is 3.7 lb. at 25 deg. F., 2.7 lb. at 35 deg. F. and 2.2 lb. at 45 deg. F. brine temperature per pound of "steam" boiled out of the evaporator.

### THE MULTIPLE EFFECT COMPRESSOR (Dual Compression)

The multiple effect compressor of Gardiner T. Voorhees is an attempt to eliminate one compressor when two evaporating temperatures are required in a plant. For reasons of high capacity and economy as regards power input it is desirable to operate a compressor at as high a suction pressure as possible, and in the case of the two temperature plant when operated with only one compressor[11] the suction pressure carried must be that for the lower temperature room. The result of this is that the other load is carried at a condition of greater piston displacement and greater power consumed than would be required under

[11] One side of the double-acting, and one cylinder of the twin vertical single-acting compressor can be used for the low temperature suction if desired.

more advantageous operating conditions. In the Voorhees device the ordinary suction stroke is taken, drawing gas from the colder evaporating coils (the lower suction pressure) and at or near the end of the stroke either ports in the cylinder will be uncovered or certain valves be operated mechanically (Fig. 46) so as to admit gas from the other coils which are operated at the higher boiling pressure.

The action of the device for securing a multiple effect compression is shown on the indicator card in Fig. 43 as well as the action of admitting the heavier pressure gas theoretically at constant volume. However, as

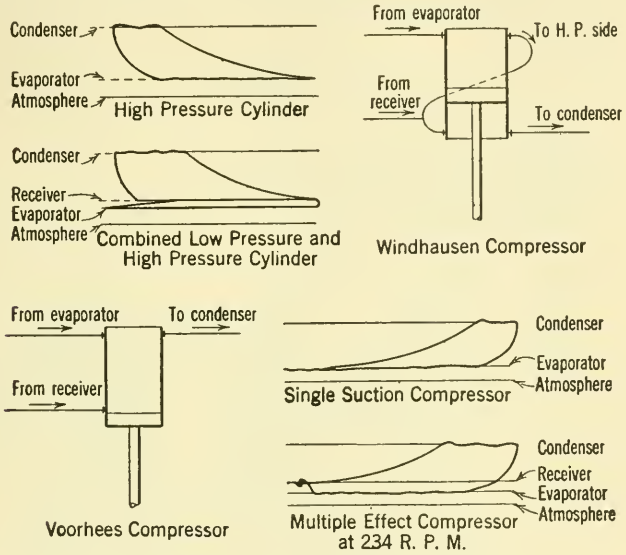



Fig. 43.—Dual (2 pressure) Compression.

the compression speed increases, this constant volume process is only approximated. The Windhausen method (Fig. 43) is not really multiple effect compression, but is more nearly stage compression. The Voorhees method is used by the Apeldoorn Company of Holland, the Seagers Company of England, and—to some slight extent—the Carbondale Machine Company and the Howe Company in the United States. As a part of the carbon dioxide compression it is of considerable value because of certain inherent failings of this refrigerant—particularly in respect to the heat of the liquid which is a large proportion of the thermal potential of the dry saturated vapor. In this case there is considerable value in permitting the refrigerant to operate under, first, a receiver pressure and, finally, the evaporating coil's pressure (Fig. 44). During the first drop of pressure, a certain amount of gas is evolved, due to the

cooling of the liquid, and this gas may be returned to the compressor at the receiver pressure instead of passing on to the evaporating coils without adequate return. For general refrigeration the multiple effect compression has not been popular because of the difficulty in adjusting the loads. In order to show the advantage to be derived *theoretically* the following problem will be solved:

**Problem.**—The condenser pressure is 185 lb. per square inch abs., suction pressure 30 and 45 lb. abs., cylinder capacity 10 cu. ft., saturated gas assumed at the beginning of the compression, no clearance in the cylinder and adiabatic compression. It is assumed that the compressor will operate under 30 lb. suction pressure in the first case, and with multiple effect in the second case. The problem

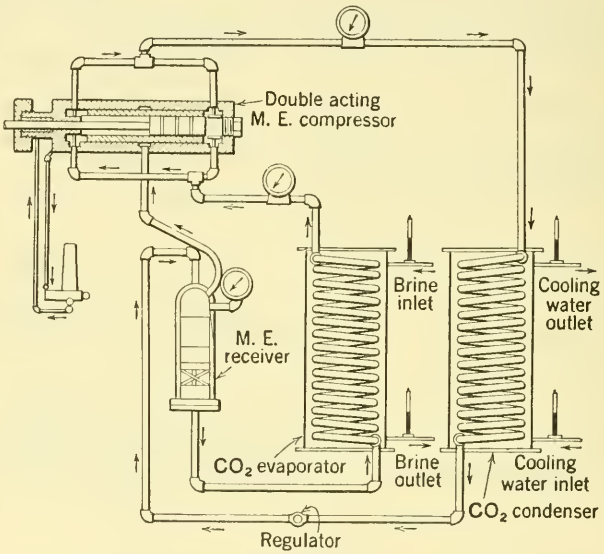



Fig. 44.—Dual Compression Applied to Carbon Dioxide Refrigeration.

is to find the percentage increase in the refrigerating capacity and the percentage increase in the power requirements for the second case. The refrigerating effect under 30 lb. suction is,

$$i''_{30\text{ lb.}} - i'_{185\text{ lb.}} = 611.4 - 145.2 = 466.2 \text{ B.t.u.}$$

$$i''_{45\text{ lb.}} - i'_{185\text{ lb.}} = 616.9 - 145.2 = 471.7 \text{ B.t.u.}$$

The weight of ammonia in 10 cu. ft. at 30 lb. abs. pressure is $\frac{10}{9.23} = 1.08$ lb.

The weight of ammonia in 10 cu. ft. at 45 lb. abs. pressure is $\frac{10}{6.3} = 1.59$ lb.

The difference, $1.59 - 1.08 = 0.51$ lb. is the amount (theoretically) of the ammonia that enters the cylinder at constant volume at the end of the suction stroke,

not making any allowance for the wire drawing which would be expected during the passage of the gas from the evaporating coils to the cylinder, and the difference in the temperatures of the two vapors. As the compressor is operated in both conditions at the same speed and the piston displacement in each case is identical, the mean effective pressure may be used to compare the power input. In the single case the refrigerating effect is the product of $1.08 \times 466.2 = 503$ B.t.u. and the mean effective pressure is 67.7 lb. per square inch. In the second case the refrigeration is $(1.08 \times 466.2) + (0.51 \times 471.7) = 743.5$ B.t.u. and the mean effective pressure is 90 lb. per square inch.

The relative advantage, then, works out as follows:

$$\text{Multiple effect compression : regular compression} = 743.7 : 503 = 1.47$$

as far as the capacity is concerned, and the power input is as $90.0 : 67.7 = 1.33$. This shows that a *gain* in the capacity of about 47 per cent is accomplished by the use of the multiple effect compression device, incidental with an increase of 33 per cent in power. This is the maximum advantage and in practice it is hardly to be expected that the full theoretical increase will be realized. If two machines had been used instead of the single two-effect compressor, one operating at 30 lb. and the other at 45 lb. suction, there would have been approximately twice the friction developed. It would seem that there are some theoretical advantages in general refrigerating practice which might disappear entirely when the conditions of operation are not the best.

In carbon dioxide compression the following problem will be of interest:

Given: The compressor is $5\frac{3}{4}$ in. by 18 in. to operate at 150 r.p.m. The piston rod is $1\frac{3}{4}$ in. and the compressor is to maintain a 25-deg. and a 40-deg. F. evaporating temperatures with 85 degrees liquefaction temperature. Taking the clearance as $4\frac{1}{2}$ per cent, the multiple effect in the cylinder as 0.625 in. diameter and the volumetric efficiency as 80 per cent, the gas entering the cylinder per minute is 59.7 cu. ft. or $\frac{59.7}{0.188}$ lb. $CO_2$. Taking the volume of the clearance space as 3.48 cu. ft., the temperature of the clearance gas as 142 deg. F. and the specific volume of this gas as 0.096 then the weight of the clearance gas is $\frac{3.48}{0.096} = 36$ lb. The total weight of gas becomes $36 + 317 = 353$ lb. of gas in the cylinder at the end of the low-pressure suction.

Taking the specific volume of the gas at 583 lb. and 48.5 deg. F. as 0.152 cu. ft. the volume occupied by this gas will be $353 \times 0.152 = 53.7$ cu. ft. The cylinder volume is 81.0 and therefore the amount of gas entering through the ports is 27.3 cu. ft. theoretically or $0.95 \times 27.3 = 25.9$ cu. ft. The weight of gas entering through the ports is $\frac{25.9}{0.152} = 170.5$ lb. per minute. Taking the net refrigeration at 25 deg. F. at 56.9 B.t.u. per pound the total refrigeration, at the low pressure, is

$$56.9 \times 317 = 18{,}000 \text{ B.t.u. per minute} = 90 \text{ tons refrigeration.}$$

Taking the net refrigeration at the higher pressure at 55.0 B.t.u. the total refrigera- is $170.5 \times 55.0 = 9400$ B.t.u. $= 47$ tons of refrigeration.

When the multiple effect is used to cool the liquid the problem is not so straightforward. In this case it is necessary to assume an intermediate evaporating temperature, and then calculate the volume obtained which must be enough for the

cylinder dimensions. Several calculations are usually necessary before the proper pressure is obtained.

## STAGE AMMONIA COMPRESSION

Stage ammonia compression is not a recent idea, but its real development did not start until about 1915. At that time the Central Cold Storage Company and the Beatrice Creamery (both of Chicago) and the Ninth Street Terminal Warehouse of Cleveland, of 250, 200 and 150 tons respectively were installed using two-stage compression for refrigerating temperatures between $-10$ and $-20$ deg. F. Since that time progress in stage compression has been very rapid, and its advantages have been brought out by papers presented before the technical societies by H. Sloan,[12] George Horne[13] and Thomas Shipley.[14] As a result it is conceded (1925) that stage compression is of advantage if the ratio of the compression $\left(\frac{p_2}{p_1}\right)$ is large enough, and as a rule that this condition prevails when the suction pressure is lowered to about 5 lb. gage. To the contrary the usual ice-making installation does not require stage compression, unless the condenser pressure is high.

Previous to 1915 the first attempt at stage compression was that of Norman Selfe who secured a patent in 1880. He used a form of trunk piston for the high-pressure piston; the design including a vertical single-acting cylinder where the compressed gas *was not* cooled between the stages. The St. Clair two-stage compressor was built later on in the eighties (Ice and Refrigeration, Jan., 1894) where water cooling was made use of between the stages. About 1890 Linde developed his two-stage compressor which also was water cooled, and his machine was used successfully in Europe and Australia. Then interest seemed to disappear because it was not clear that stage compression offered sufficient advantages to offset the many disadvantages due to increased fixed charges and the more complicated compressor. When there was a demand for low-temperature refrigeration the absorption machine appeared to be able to carry this kind of load with satisfaction and economy of performance, especially as steam was used universally to provide power for the refrigerating machine up to about 1915.

Occasionally the *booster compressor* was used when a relatively small part of the load required a low temperature. These booster compressors took the vapor from the low-pressure coils and raised the pressure

[12] H. Sloan, Amer. Soc. Refrigerating Eng., Dec., 1916.

[13] George Horne, Amer. Soc. Refrigerating Eng., May, 1922.

[14] Thomas Shipley, National Assoc. Practical Refrig. Eng., Nov., 1924.

to that of the other coils so that the standard compressor could handle all of the gas and discharge it all into the condenser. The booster was very convenient, and served its purpose in a satisfactory manner. However, stage ammonia compression is a problem by itself and much more complicated than in the case of the compression of air.

Stage compression for ammonia is different from that of air because

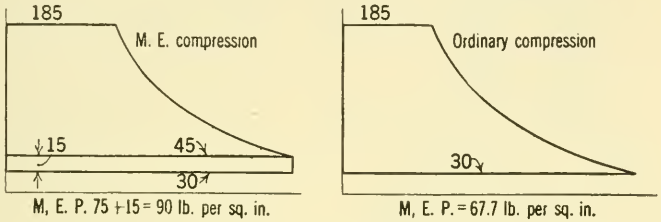



Fig. 45.—Theoretical Indicator Diagrams for Dual Compression.

ammonia in a part of the cycle is a liquid, and in addition the temperatures carried in part of the cycle are frequently 100 degrees below the temperature of the atmosphere. Also, unlike air compression, the pressure range varies very decidedly with the seasons for in the winter time the pressure ratio may be as small as 6.0 whereas in the summer this ratio of the discharge to the suction pressure may be increased to 12.0. The air compressor is likely to be operated for the same discharge pressures summer and winter.

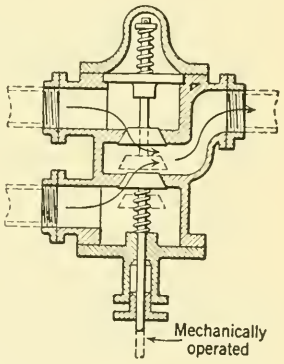



Fig. 46.—Dual Compression Mechanically Operated Valve.

In the refrigerating cycle there is an evolution of gas during the pressure drop incidental to the passage through the expansion valves. This gas is formed by the vaporization of some of the liquid ammonia and may amount to from 6 to 15 per cent of the liquid passing the valve, the amount depending on the initial and the final temperatures of the liquid during the passage of the expansion valve. In stage ammonia compression it has been the usual custom to use two *pressure-reducing* valves, and to permit the gas evolved during the first pressure drop to be separated from the liquid and to pass immediately into the suction line to the high-pressure cylinder. There are two reasons for this method; first the gas has a negligible amount of value as a refrigerating medium, and secondly, permitting it to pass into the low-pressure coils increases in proportion

the amount of gas to be handled by the low-pressure cylinder, and increases also the work performed by the compressor. The specific volume of ammonia assumes very large proportions as the pressure is reduced, and in the following problem the specific volume (the volume of one pound) is 16.66 and the per cent of the ammonia gasified during the first pressure reduction is 9.16 per cent—point *8* in the figure. If this gas is allowed to get into the low-pressure coils then the low-pressure cylinder would have to pump it up again to the intermediate pressure, the pressure of 53.7 lb. in the problem, and there would be little useful refrigeration to offset the expense of the process. The work done in the high-pressure cylinder would be the same in every case except for the superheat in the gas entering the high-pressure cylinder, which affects both the volume of the fluid to be compressed and therefore the amount of the work done in the cylinder.

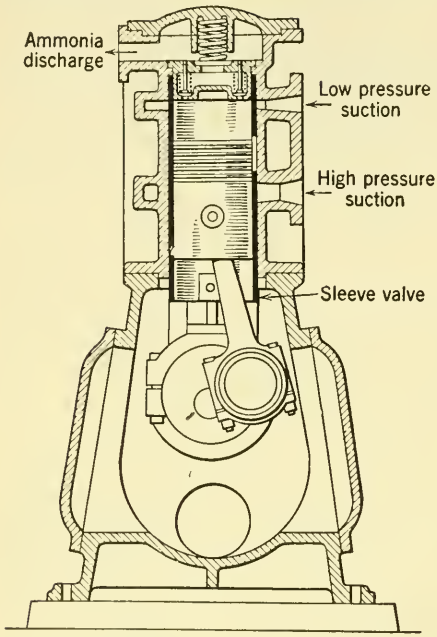



FIG. 47.—The West Sleeve Type [mechanically operated] Ammonia Compressor.

Different cycles of operation are used in stage compression depending on the method of handling the gas after it is discharged from the low-pressure cylinder. The air compressor—dealing as it does with temperatures above that of the atmosphere—attempts to cool the discharged gas from the low pressure to the initial temperature at suction. This is very easily done by the use of water-cooling coils. In ammonia compression, in the illustrative problem, the temperature of saturation at the intermediate pressure is 25 deg. F. which is considerably cooler than any water which would be available for the purpose of intermediate cooling. In fact, if cooling down to 25 degrees is required, the only way that it can be done is to make use of ammonia, but ammonia at the pressure of the low pressure, giving − 25 deg. F. boiling temperature, would not be an economical medium of operation, and one would not think for a moment of trying to cool this gas down to − 25 degrees as would be done in air compression.

It is clear therefore that cooling back to the isothermal is out of the question for ammonia compression. The very best that can be done in the intercooler is to cool down to the temperature of saturation at the intercooler pressure, and this may be done by (*a*) cooling entirely by means of liquid ammonia, by permitting the gas discharged from the low-pressure cylinder to pass into a sort of accumulator where the superheated gas and the liquid refrigerant will mix and by (*b*) cooling the superheated gas by water to (say) 70 deg. F. and by allowing this gas now superheated 45 degrees in the problem to pass into a second cooler where liquid ammonia at 25 degrees is held. The superheated gas will

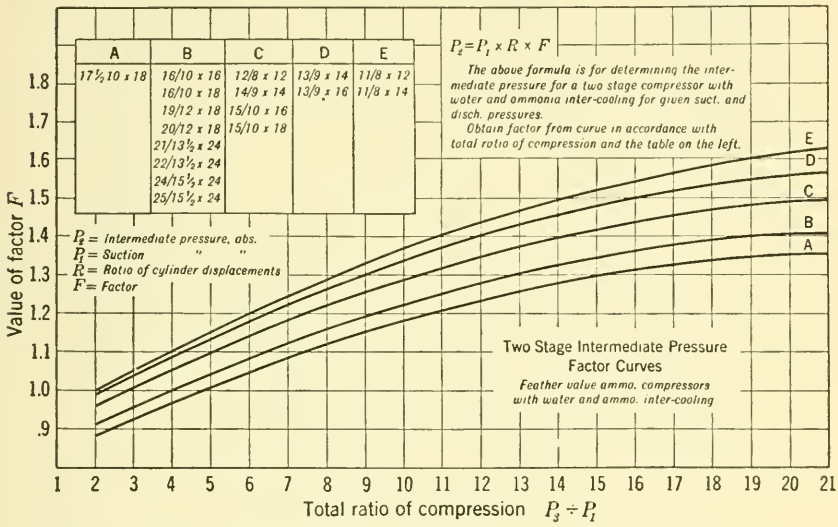

A
B
C
D
E
17½ 10 x 18
16/10 x 16
16/10 x 18
19/12 x 18
20/12 x 18
21/13½ x 24
22/13½ x 24
24/15½ x 24
25/15½ x 24
12/8 x 12
14/9 x 14
15/10 x 16
15/10 x 18
13/9 x 14
13/9 x 16
11/8 x 12
11/8 x 14
$P_2 = P_1 \times R \times F$
The above formula is for determining the intermediate pressure for a two stage compressor with water and ammonia inter-cooling for given suct. and disch. pressures.
Obtain factor from curve in accordance with total ratio of compression and the table on the left.
$P_2$ = Intermediate pressure, abs.
$P_1$ = Suction " "
$R$ = Ratio of cylinder displacements
$F$ = Factor
Two Stage Intermediate Pressure Factor Curves
Feather value ammo. compressors with water and ammo. inter-cooling
Value of factor F
Total ratio of compression $P_3 \div P_1$

Fig. 48.—Two-stage Compression; Intermediate Pressure Determination.

be cooled, and some of the liquid will be evaporated as it absorbs heat and boils at 25 deg. F. The illustrative problem also assumes a case (*c*) where the second process (*b*) is omitted and gas superheated 45 degrees is permitted to pass on to the high-pressure cylinder.

Lately there has been a tendency on the part of a few persons to try to simplify the stage compression cycle. This has been directed against the use of the accumulator (the intercooler) and towards the use of only one expansion valve with cooling between the stages by water only. In the illustrative example the temperature after compression in the low-pressure cylinder up to 53.7 lb. is about 107 deg. F. and cooling in the aftercooler with the use of water would be from 107 to 70 degrees. This temperature of the gas entering the high-pressure cylinder would not be excessive provided the condenser pressure was not very high. In the illustrative example (Fig. 49) it will be seen

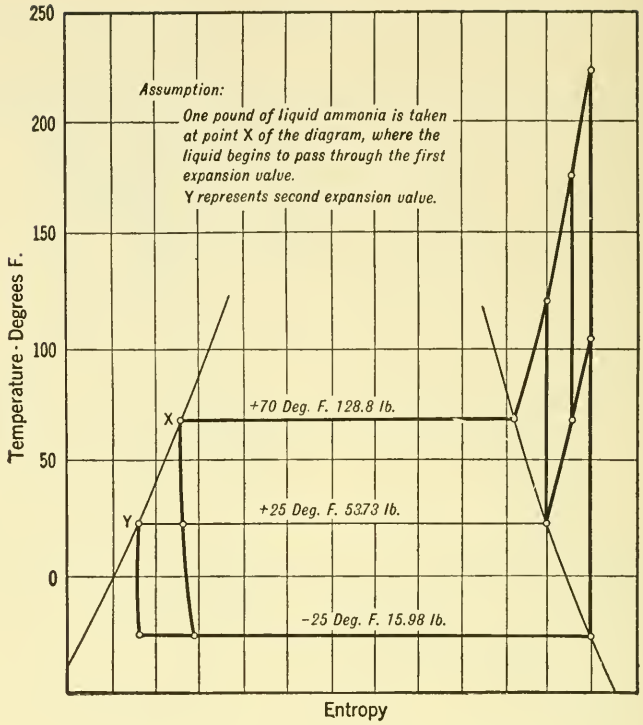



FIG. 49.—Simple and Stage Compression.

| Problem | Case A<br>Cool to +25 Deg. F. with Liquid $NH^3$ | Case B<br>Cool to 70 Deg. F. with Water and to +25 Deg. F. with $NH^3$ | Case C<br>Cool to 70 Deg. F. with Water | Case D<br>Simple Compression | Case E<br>No Liquid Intercooling Gas Cooled to 70 Deg. F. with Water |
|---|---|---|---|---|---|
| Weight of liquid evaporated in cooling l.p. disch. gas, lb. | 0.0725 | 0.04275 | | | |
| Weight liquid evaporated in first exp. valve, lb. | 0.0916 | 0.0916 | 0.0916 | 0.178 | 0.0916 |
| Weight of liquid passing to second expansion, lb. | 0.8359 | 0.8657 | 0.9084 | | |
| Weight ammonia as liquid available for refrigeration, lb. | 0.7588 | 0.7860 | 0.822 | 0.822 | 0.822 |
| Refrigerating effect, B.t.u. | 445.50 | 461.0 | 484.2 | 482.7 | 482.7 |
| Work done—l.p.cyl. ft., lb. | 44,450 | 46,020 | 48,300 | ..... | 53,150 |
| Work done—h.p. cyl. ft.-lb. | 39,800 | 39,800 | 44,450 | ..... | 44,450 |
| Total work of compression, ft.-lb. | 84,250 | 85,820 | 92,750 | 101,400 | 97,600 |
| Probable vol. eff. h.p. cyl. | 0.9233 | 0.9233 | 0.9165 | 0.8110 | 0.9165 |
| Probable eff. l.p. cyl. | 0.9008 | 0.9008 | 0.9008 | ..... | 0.9008 |
| Total work—allowing for vol. eff. ft.-lb. | 92,450 | 94,250 | 102,100 | 125,000 | 107,500 |
| Coefficient of performance | 3.75 | 3.81 | 3.690 | 3.00 | 3.49 |
| Hp. per ton of refrigeration | 1.257 | 1.238 | 1.277 | 1.568 | 1.352 |

Comparative Efficiencies of Simple and Compound Compression.

that this case—Case E—is not so good as far as the horse power per ton of refrigeration is concerned. Referring to Cases A, B and C it will be noticed that there is slight difference in this respect, but that in Case B the lowest horse power per ton of refrigeration is found as developed from the theoretical calculation. Finally there are given in the table values calculated for simple compression, Case D. All the calculations are based on the premise of having one pound of liquid at 70 deg. F. at the expansion valve.

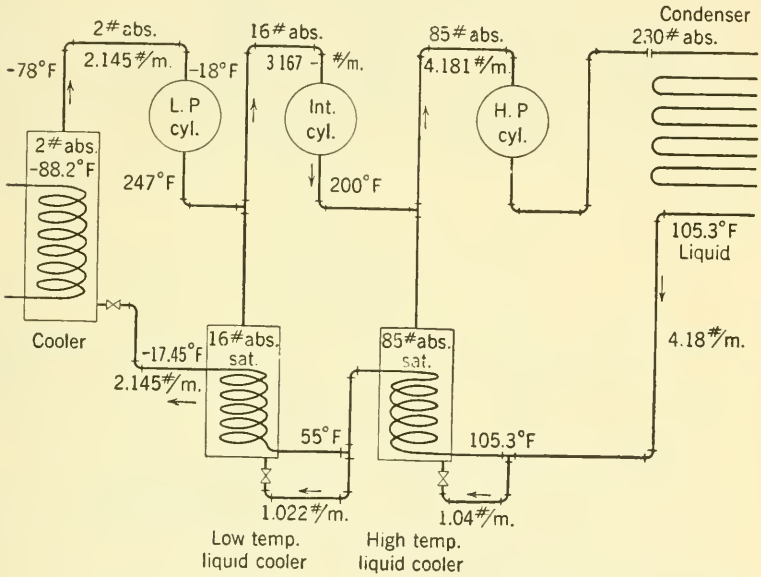



Fig. 50.—Ammonia Compression in Three Stages.

In general it can be said that the best arrangement is usually the simplest, but this is not the case when dealing with low-temperature refrigeration. Ordinarily there is no advantage in cooling the refrigerant by means of itself, although that is what is done in the case when using the accumulator in the flooded system. Referring to the table it will be seen that the simplest cycle for stage compression, Case D, is not the best one, as far as the power requirements are concerned, by any means, although it has one less piece of equipment, the accumulator. With the accumulator and cooling only to 70 deg. F. with water the horse power per ton of refrigeration is 1.277 as compared with 1.352 in the

cycle in Case E. The best results are given in Case B which has the accumulator, two expansion valves and has cooling with water to 70 degrees and with ammonia to 25 degrees.

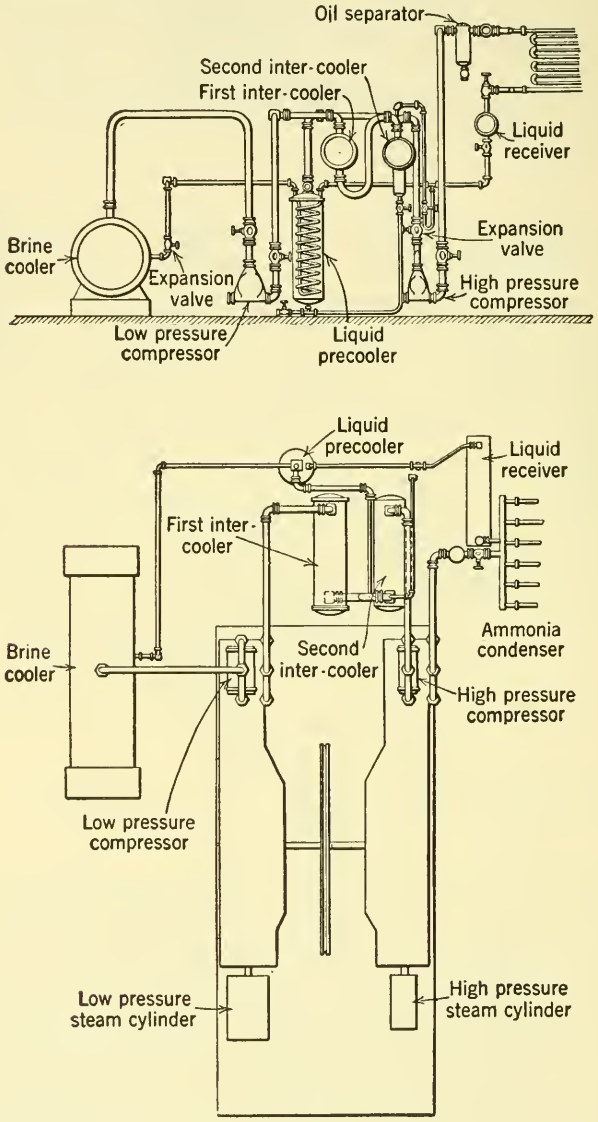



Fig. 51.—Two-stage Ammonia Compressor.

Ammonia compression in three stages is rare at the present time, as only extremely severe operating conditions will warrant such a design.

A Carbondale three-stage compressor (Fig. 50) installed in one of the rubber manufacturing companies had a 14 and 8 and 4 by 12 inch stroke,

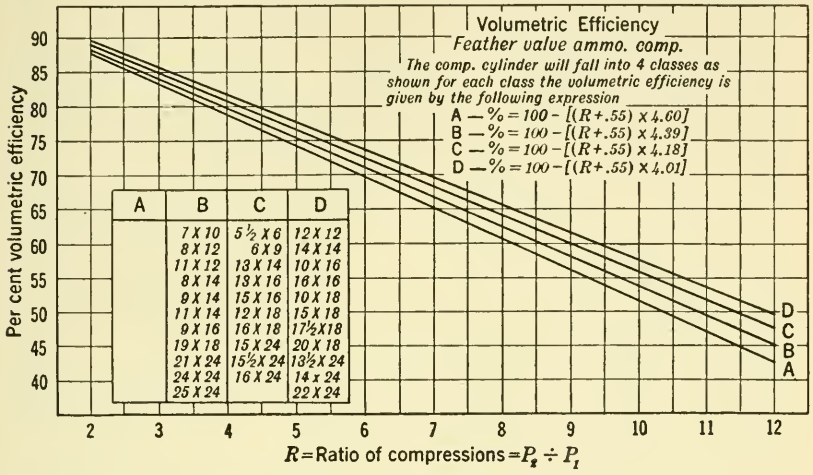

Volumetric Efficiency
Feather valve ammo. comp.
The comp. cylinder will fall into 4 classes as shown for each class the volumetric efficiency is given by the following expression
A — % = 100 − [(R + .55) × 4.60]
B — % = 100 − [(R + .55) × 4.39]
C — % = 100 − [(R + .55) × 4.18]
D — % = 100 − [(R + .55) × 4.01]
A | B | C | D
7 X 10 | 5½ X 6 | 12 X 12
8 X 12 | 6 X 9 | 14 X 14
11 X 12 | 13 X 14 | 10 X 16
8 X 14 | 13 X 16 | 16 X 16
9 X 14 | 15 X 16 | 10 X 18
11 X 14 | 12 X 18 | 15 X 18
9 X 16 | 16 X 18 | 17½X18
19 X 18 | 15 X 24 | 20 X 18
21 X 24 | 15½X 24 | 13½X 24
24 X 24 | 16 X 24 | 14 x 24
25 X 24 | | 22 X 24
Per cent volumetric efficiency
R = Ratio of compressions = $P_2 \div P_1$
D
C
B
A

Fig. 52.—Volumetric Efficiency.

single-acting except for the low pressure (14 by 12) cylinder which was double-acting. The total piston displacement was 570 cu. ft. per minute at 225 r.p.m. and the operating conditions were for 2.0 lb. suction and

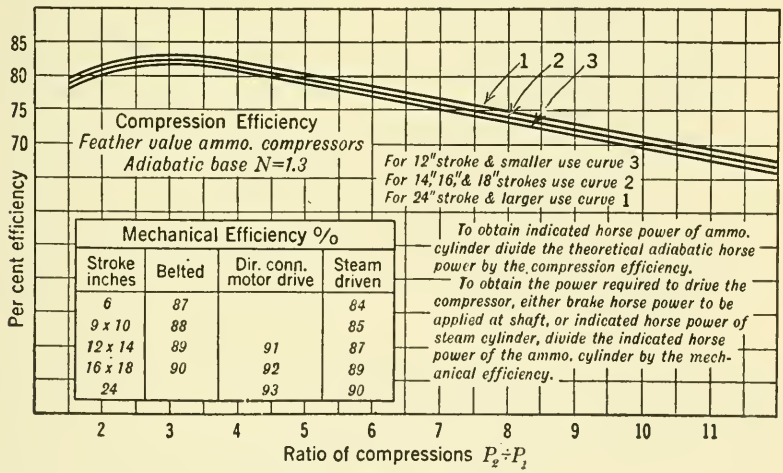

Compression Efficiency
Feather valve ammo. compressors
Adiabatic base N=1.3
For 12" stroke & smaller use curve 3
For 14", 16" & 18" strokes use curve 2
For 24" stroke & larger use curve 1
1
2
3
Mechanical Efficiency %
Stroke inches | Belted | Dir. conn. motor drive | Steam driven
6 | 87 | | 84
9 x 10 | 88 | | 85
12 x 14 | 89 | 91 | 87
16 x 18 | 90 | 92 | 89
24 | | 93 | 90
To obtain indicated horse power of ammo. cylinder divide the theoretical adiabatic horse power by the compression efficiency.
To obtain the power required to drive the compressor, either brake horse power to be applied at shaft, or indicated horse power of steam cylinder, divide the indicated horse power of the ammo. cylinder by the mechanical efficiency.
Per cent efficiency
Ratio of compressions $P_2 \div P_1$

Fig. 53.—Compression Efficiency.

230 lb. condenser pressure. This machine was designed for 6.0 tons under good conditions of operation and would require about 30 brake horse power.

**Compressor Details.**—Tables 5 to 13 give approximate sizes of standard ammonia and carbonic compressors built by various manufacturers of refrigerating machines. In addition Figs. 52, 53 and 54 give the volumetric efficiency, cubic feet per ton and horse power per ton used by a well-known manufacturer. Every manufacturer has curves similar to these last, and the only variations are in the volumetric efficiency and the allowance for the leakage through the valves and piston rings and in the imperfections of operation. In Figs. 29 and 30, for example, no allowance was made for mechanical defects nor for the personal equation of the operator so that where the values given in Figs. 29 and 30 are used it is necessary to make some allowance as a factor of safety. Likewise the horse power per ton of refrigeration as given in Figs. 31 and 32 are smaller than would be specified in practice. In starting up, especially in automatic installations, the load is considerably greater than under usual running conditions because of the high suction pressures, and so the result is that a larger motor is installed than would be indicated from the consideration of Figs. 31 and 32. Tables 3 to 14 are taken from manufacturer's data without editing.

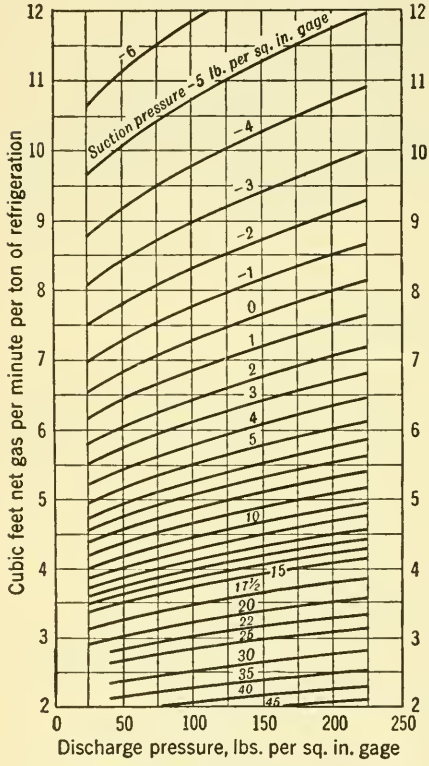
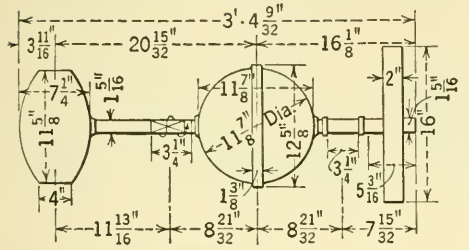



Fig. 54.—Piston Displacement per Ton of Refrigeration per Minute.

In the tables the floor space and in a number of cases the overall height are given. Piping connections are given in Chapter IV, as are also the details on suction traps and oil separators. Some electric motor characteristics are shown in Chapter XXII, and some steam engine details are collected in Chapter XXIII.

## TABLE 2

### Characteristics of No. 2 Audiffren-Singrun Refrigerating Machine




Direction of rotation is clockwise when facing machine at pulley end

DUMBBELL AND PULLEY

Dumbbell Type 2E

Keyway in shaft....$\frac{1}{4}$ in. by $\frac{1}{8}$ in. by $4\frac{3}{4}$ in.

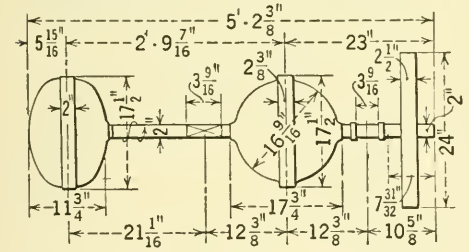
Weight.....140 lb.

Weight crated.....200 lb.

R.p.m.....380

| Condensing Water Temperature at Overflow | | 50° | 59° | 68° | 77° | 86° | 95° |
|---|---|---|---|---|---|---|---|
| Capacity in B.t.u. absorbed per hour at brine temperature of..... | 50° | 4200 | 4000 | 3800 | 3640 | 3450 | 3250 |
| | 41° | 3560 | 3400 | 3250 | 3090 | 2930 | 2730 |
| | 37° | 3290 | 3170 | 3010 | 2850 | 2690 | 2530 |
| | 32° | 3050 | 2890 | 2770 | 2610 | 2490 | 2340 |
| | 27° | 2610 | 2500 | 2380 | 2260 | 2140 | 2020 |
| | 23° | 2410 | 2340 | 2220 | 2100 | 1980 | 1860 |
| | 14° | 1940 | 1820 | 1740 | 1660 | 1580 | 1460 |
| Capacity in pounds of ice per hour with water for cans at 59 degrees and brine at 27 degrees..... | | 13 | 12 | 11 | 10½ | 10 | 9 |
| Condensing water required per hour in gallons with temperature rise of.... | 9° | 57 | 53 | 49 | 45 | 41 | 38 |
| | 18° | 28.5 | 26.5 | 24.5 | 22.5 | 20.5 | 19 |
| | 27° | ..... | 18 | 16.5 | 15 | 13.5 | 13 |
| | 36° | ..... | ..... | 12.5 | 11.5 | 10.5 | 9.5 |
| With brine at 27 deg. F..... | | | | | | | |
| Horse power required brine at 27½ deg. F..... | | .48 | .50 | .53 | .55 | .58 | .60 |

### Characteristics of No. 3 Audiffren-Singrun Refrigerating Machine




Direction of rotation is clockwise when facing machine at pulley end

DUMBBELL AND PULLEY

Dumbbell Type 3E

Keyway in shaft....$\frac{3}{8}$ in. by $\frac{3}{16}$ in. by $7\frac{5}{16}$ in.

Weight.....390 lb.

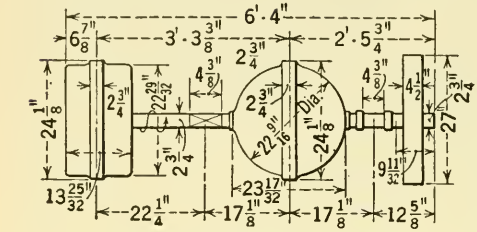
Weight crated.....490 lb.

R.p.m.....280

| Condensing Water Temperature at Overflow | | 50° | 59° | 68° | 77° | 86° | 95° |
|---|---|---|---|---|---|---|---|
| Capacity in B.t.u. absorbed per hour at brine temperature of..... | 50° | 10,500 | 10,000 | 9500 | 9100 | 8610 | 8120 |
| | 41° | 8,910 | 8,510 | 8120 | 7720 | 7320 | 6830 |
| | 37° | 8,220 | 7,920 | 7520 | 7120 | 6730 | 6340 |
| | 32° | 7,620 | 7,230 | 6930 | 6530 | 6240 | 5840 |
| | 27° | 6,530 | 6,240 | 5940 | 5640 | 5340 | 5040 |
| | 23° | 6,040 | 5,840 | 5540 | 5250 | 4950 | 4650 |
| | 14° | 4,850 | 4,550 | 4350 | 4160 | 3960 | 3860 |
| Capacity in pounds of ice per hour with water for cans at 59 degrees and brine at 27 degrees..... | | 33 | 30 | 27 | 26 | 24 | 23 |
| Condensing water required per hour in gallons with temperature rise of.... | 9° | 120 | 114 | 107 | 100 | 93 | 85 |
| | 18° | 60 | 57 | 54 | 50 | 47 | 43 |
| | 27° | ..... | 38 | 36 | 33 | 31 | 28 |
| | 36° | ..... | ..... | 27 | 25 | 24 | 22 |
| With brine at 27 deg. F..... | | | | | | | |
| Horse power required brine at 27½ deg. F..... | | .90 | 1.02 | 1.14 | 1.26 | 1.38 | 1.50 |

TABLE 2—*Continued*

CHARACTERISTICS OF NO. 4 AUDIFFREN-SINGRUN REFRIGERATING MACHINE




Direction of rotation is clockwise when facing machine at pulley end

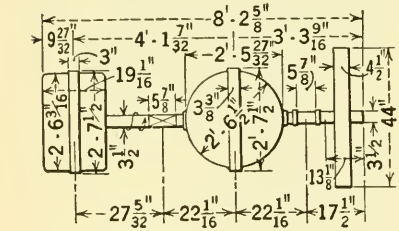
DUMBBELL AND PULLEY

DUMBBELL TYPE 4H

Keyway in shaft.....½ in. by ¼ in. by 8¾ in.
Weight.....850 lb.
Weight crated.....1085 lb.
R.p.m.....190

| Condensing Water Temperature at Overflow | | 50° | 59° | 68° | 77° | 86° | 95° |
|---|---|---|---|---|---|---|---|
| Capacity in B.t.u. absorbed per hour at brine temperature of..... | 50° | 21,000 | 20,000 | 19,000 | 18,200 | 17,220 | 16,240 |
| | 41° | 17,820 | 17,020 | 16,240 | 15,440 | 14,640 | 13,660 |
| | 37° | 16,440 | 15,840 | 15,040 | 14,240 | 13,460 | 12,680 |
| | 32° | 15,240 | 14,460 | 13,860 | 13,060 | 12,480 | 11,680 |
| | 27° | 13,060 | 12,480 | 11,880 | 11,280 | 10,680 | 10,100 |
| | 23° | 12,080 | 11,680 | 11,080 | 10,500 | 9,900 | 9,300 |
| | 14° | 9,700 | 9,100 | 8,700 | 8,320 | 7,920 | 7,720 |
| Capacity in pounds of ice per hour with water for cans at 59 degrees and brine at 27 degrees..... | | 66 | 60 | 55 | 52 | 49 | 45 |
| Condensing water required per hour in gallons with temperature rise of.... | 9° | 230 | 220 | 210 | 195 | 185 | 175 |
| | 18° | 115 | 110 | 105 | 97 | 93 | 88 |
| | 27° | ..... | 74 | 70 | 65 | 62 | 58 |
| | 36° | ..... | ..... | 53 | 49 | 47 | 44 |
| With brine at 27 deg. F..... | | | | | | | |
| Horse power required brine at 27½ deg. F.... | | 1.6 | 1.8 | 2.05 | 2.15 | 2.3 | 2.5 |

CHARACTERISTICS OF NO. 6 AUDIFFREN-SINGRUN REFRIGERATING MACHINE




Direction of rotation is clockwise when facing machine at pulley end

DUMBBELL AND PULLEY

DUMBBELL TYPE 6H

Keyway in shaft.....½ in. by ¼ in. by 12¾ in.
Weight.....1700 lb.
Weight crated.....2100 lb.
R.p.m.....140

| Condensing Water Temperature at Overflow | | 50° | 59° | 68° | 77° | 86° | 95° |
|---|---|---|---|---|---|---|---|
| Capacity in B.t.u. absorbed per hour at brine temperature of..... | 50° | 42,000 | 40,000 | 38,000 | 36,400 | 34,490 | 32,480 |
| | 41° | 35,640 | 34,040 | 32,480 | 30,880 | 29,280 | 27,320 |
| | 37° | 32,880 | 31,680 | 30,080 | 28,480 | 26,820 | 25,360 |
| | 32° | 30,480 | 28,920 | 27,720 | 26,120 | 24,960 | 23,360 |
| | 27° | 26,120 | 24,960 | 23,760 | 22,560 | 21,360 | 20,200 |
| | 23° | 24,160 | 23,360 | 22,160 | 21,000 | 19,800 | 18,600 |
| | 14° | 19,400 | 18,200 | 17,400 | 16,640 | 15,840 | 15,440 |
| Capacity in pounds of ice per hour with water for cans at 59 degrees and brine at 27 degrees..... | | 132 | 120 | 110 | 105 | 99 | 92 |
| Condensing water required per hour in gallons with temperature rise of.... | 9° | 420 | 410 | 400 | 390 | 380 | 370 |
| | 18° | 210 | 205 | 200 | 195 | 190 | 185 |
| | 27° | ..... | 137 | 134 | 130 | 127 | 124 |
| | 36° | ..... | ..... | 100 | 95 | 93 | 92 |
| With brine at 27 deg. F..... | | | | | | | |
| Horse power required brine at 27½ deg. F.... | | 3.6 | 3.9 | 4.1 | 4.3 | 4.6 | 4.8 |

## TABLE 3

### Transmission Data for Belt-driven V. S. A. Enclosed Type Compressors

| Size of Machine, Inches | Standard Speed, R.p.m. | Diameter of Belt Wheel, Inches | Face of Belt Wheel (Not Including Poekets), Inches | Weight of Belt Wheel, Pounds | Motor, Full Load, R.p.m., 60 Cycles | Diameter of Motor Pulley, Inches | Face of Motor Pulley, Inches | Width of Double Load Belt, Inches | Width of Single Load Belt, Inches | Distance, Center to Center, of Belt Wheel and Motor Pulley, Feet | Approximate Length of Belt, Feet |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 3× 3 (Single) | 300 | 26 | 3½ | 180 | 1720 | 4½ | 3½ | ..... | 3 | 6 | 17 |
| 3× 3 (Double) | 300 | 26 | 3½ | 180 | 1720 | 4½ | 3½ | ..... | 3 | 6 | 17 |
| 4× 4 (Single) | 275 | 30 | 4½ | 300 | 1720 | 5 | 4½ | ..... | 4 | 6 | 18 |
| 4× 4 (Double) | 275 | 30 | 4½ | 300 | 1720 | 5 | 4½ | ..... | 4 | 6 | 18 |
| 5× 5 | 240 | 36 | 6½ | 600 | 1150 | 7½ | 6½ | ..... | 6 | 8 | 22 |
| 6× 6 | 220 | 42 | 7½ | 600 | 1150 | 8 | 7½ | ..... | 7 | 8 | 24 |
| 7× 7 | 210 | 48 | 8½ | 1000 | 1150 | 9 | 8½ | 6 | 8 | 10 | 29 |
| 8× 8 | 200 | 50 | 8½ | 1200 | 870 | 11½ | 8½ | 8 | ..... | 10 | 30 |
| 9× 9 | 190 | 60 | 10½ | 1500 | 870 | 13 | 10½ | 10 | ..... | 12 | 34 |
| 10×10 | 180 | 72 | 12½ | 1800 | 870 | 15 | 12½ | 12 | ..... | 14 | 40 |
| 12×12 | 170 | 86 | 14½ | 3250 | 870 | 17 | 14½ | 14 | ..... | 16 | 47 |

Weight of belt wheels based on 225-lb. H. P., and 25-lb. B. P. Standard Speed.

Compressors requiring 20 per cent or more speed reduction will require special belt wheels of greater weight.

## TABLE 4

### Motor Engine Horse Power for V. S. A. Enclosed Compressors

185 Lb. Condenser Pressure

| Cylinder | | | Revolutions per minute | Ammonia Suction Pressures and Corresponding Ammonia Temperatures, Pounds per Square Inch, Gage and Degrees Fahrenheit | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | 6 Lb. −15 deg. F. | | 9 Lb. −10 deg. F. | | 12 Lb. −5 deg. F. | | 16 Lb. −0 deg. F. | |
| Number | Bore | Stroke | | E.H.P. | M.H.P. | E.H.P. | M.H.P. | E.H.P. | M.H.P. | E.H.P. | M.H.P. |
| 1 | 3 | 3 | 330 | 1.35 | 1.22 | 1.43 | 1.29 | 1.51 | 1.36 | 1.58 | 1.43 |
| 1 | 4 | 4 | 275 | 2.85 | 2.60 | 3.01 | 2.8 | 3.18 | 2.9 | 3.32 | 3.10 |
| 2 | 4 | 4 | 275 | 5.46 | 5.10 | 5.80 | 5.40 | 6.10 | 5.70 | 6.38 | 6.00 |
| 2 | 5 | 5 | 240 | 9.10 | 8.60 | 9.60 | 9.20 | 10.1 | 9.7 | 10.6 | 10.1 |
| 2 | 6 | 6 | 220 | 14.2 | 13.7 | 15.1 | 14.4 | 15.9 | 15.2 | 16.6 | 15.9 |
| 2 | 7 | 7 | 210 | 21.2 | 20.5 | 22.5 | 21.6 | 23.7 | 22.8 | 24.8 | 23.9 |
| 2 | 8 | 8 | 200 | 29.9 | 28.9 | 31.6 | 30.6 | 33.3 | 32.2 | 34.8 | 33.8 |
| 2 | 9 | 9 | 190 | 40.5 | 38.8 | 42.8 | 41.0 | 45.2 | 43.4 | 47.3 | 45.3 |
| 2 | 10 | 10 | 180 | 52.5 | 49.9 | 55.5 | 52.8 | 58.5 | 55.8 | 61.3 | 58.3 |
| 2 | 12 | 12 | 170 | 86.0 | 80.7 | 90.8 | 85.3 | 96.0 | 90.3 | 100.2 | 94.2 |
| | | | | 19 Lb. 5 deg. F. | | 24 Lb. 10 deg. F. | | 28 Lb. 15 deg. F. | | 33 Lb. 20 deg. F. | |
| | | | | E.H.P. | M.H.P. | E.H.P. | M.H.P. | E.H.P. | M.H.P. | E.H.P. | M.H.P. |
| 1 | 3 | 3 | 330 | 1.65 | 1.50 | 1.70 | 1.54 | 1.76 | 1.59 | 1.80 | 1.63 |
| 1 | 4 | 4 | 275 | 3.47 | 3.20 | 3.60 | 3.30 | 3.71 | 3.40 | 3.80 | 3.50 |
| 2 | 4 | 4 | 275 | 6.65 | 6.30 | 6.90 | 6.50 | 7.10 | 6.70 | 7.30 | 6.80 |
| 2 | 5 | 5 | 240 | 11.1 | 10.7 | 11.5 | 11.0 | 11.9 | 11.3 | 12.1 | 11.6 |
| 2 | 6 | 6 | 220 | 17.4 | 16.6 | 18.0 | 17.2 | 18.6 | 17.8 | 19.0 | 18.2 |
| 2 | 7 | 7 | 210 | 25.9 | 25.0 | 26.8 | 25.8 | 27.7 | 26.8 | 28.3 | 27.2 |
| 2 | 8 | 8 | 200 | 36.4 | 35.3 | 37.7 | 36.3 | 38.9 | 37.7 | 39.8 | 38.4 |
| 2 | 9 | 9 | 190 | 48.4 | 47.4 | 51.1 | 48.9 | 52.8 | 50.5 | 54.0 | 51.7 |
| 2 | 10 | 10 | 180 | 64.0 | 61.0 | 66.2 | 63.0 | 68.5 | 65.3 | 70.0 | 66.4 |
| 2 | 12 | 12 | 170 | 105.0 | 98.4 | 108.4 | 102.0 | 112.0 | 105.4 | 114.5 | 107.6 |

E.H.P. = Indicated horse power of direct compressor steam engine required.
M.H.P. = Motor horse power required.

### Multiply above Horse Power by Constants Given below to Obtain Horse Power at Different Condenser Pressures Given

| Condensed Pressures, Pounds, Gage | Ammonia Suction Pressures | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | 6 Lb. | 9 Lb. | 12 Lb. | 16 Lb. | 19 Lb. | 24 Lb. | 28 Lb. | 33 Lb. |
| 200 | 1.042 | 1.048 | 1.05 | 1.052 | 1.057 | 1.06 | 1.062 | 1.07 |
| 168 | .85 | .948 | .945 | .943 | .94 | .938 | .93 | .928 |
| 153 | .8895 | .8893 | .8885 | .888 | .8872 | .8868 | .886 | .8847 |

TABLE 4—*Continued*

VERTICAL SINGLE-ACTING MACHINES

Volumetric Efficiencies of Compressors—Efficiency. Actual Cubic Feet Piston Displacement per Ton per Minute—Cubic Feet. Indicated Horse Power of Compressors per Ton of Refrigeration—Horse Power.

| Compressor Pressure (Gage) | Corresponding Temperature, Degrees | Back Pressure (Gage) and Corresponding Temperature | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 7.94 Lb. −12 Degrees | | | 10.31 Lb. −80 Degrees | | | 12.89 Lb. −40 Degrees | | | 15.67 Lb. −0 Degrees | | |
| | | EFF. | C.F. | H.P. | EFF. | C.F. | H.P. | EFF. | C.F. | H.P. | EFF. | C.F. | H.P. |
| 126.20 | 75 | 79.0 | 6.32 | 1.38 | 80.2 | 5.83 | 1.30 | 81.4 | 5.13 | 1.16 | 82.6 | 4.47 | 1.05 |
| 139.40 | 80 | 78.5 | 6.50 | 1.48 | 79.7 | 5.89 | 1.39 | 80.8 | 5.21 | 1.25 | 81.9 | 4.55 | 1.13 |
| 153.18 | 85 | 78.0 | 6.60 | 1.58 | 79.1 | 5.95 | 1.48 | 80.2 | 5.30 | 1.34 | 81.3 | 4.64 | 1.24 |
| 167.92 | 90 | 77.5 | 6.70 | 1.69 | 78.6 | 6.05 | 1.59 | 79.6 | 5.40 | 1.46 | 80.6 | 4.72 | 1.34 |
| 183.65 | 95 | 77.0 | 6.79 | 1.83 | 78.0 | 6.12 | 1.68 | 79.0 | 5.50 | 1.56 | 80.0 | 4.81 | 1.45 |
| 200.42 | 100 | 76.5 | 6.93 | 1.96 | 77.5 | 6.20 | 1.82 | 78.4 | 5.60 | 1.67 | 79.4 | 4.90 | 1.56 |
| 218.28 | 105 | 76.0 | 7.05 | 2.10 | 76.9 | 6.30 | 1.93 | 77.8 | 5.71 | 1.79 | 78.7 | 4.99 | 1.64 |
| 237.27 | 110 | 75.5 | 7.18 | 2.22 | 76.4 | 6.40 | 2.04 | 77.2 | 5.80 | 1.90 | 78.1 | 5.10 | 1.74 |
| 258.70 | 115 | 75.0 | 7.35 | 2.37 | 75.8 | 6.50 | 2.16 | 76.6 | 5.90 | 2.01 | 77.5 | 5.20 | 1.84 |
| 275.90 | 120 | 74.5 | 7.50 | 2.52 | 75.3 | 6.63 | 2.31 | 76.0 | 6.02 | 2.13 | 76.8 | 5.30 | 1.95 |
| | | 19.46 Lb. +5 Degrees | | | 23.64 Lb. +10 Degrees | | | 28.24 Lb. +15 Degrees | | | 33.25 Lb. +20 Degrees | | |
| | | EFF. | C.F. | H.P. | EFF. | C.F. | H.P. | EFF. | C.F. | H.P. | EFF. | C.F. | H.P. |
| 126.20 | 75 | 83.7 | 3.99 | .96 | 84.8 | 3.49 | .86 | 85.9 | 3.07 | .77 | 86.9 | 2.70 | .60 |
| 139.40 | 80 | 83.0 | 4.05 | 1.00 | 84.1 | 3.56 | .95 | 85.2 | 3.13 | .85 | 86.2 | 2.75 | .76 |
| 153.18 | 85 | 82.3 | 4.14 | 1.14 | 83.4 | 3.62 | 1.03 | 84.4 | 3.20 | .95 | 85.5 | 2.81 | .85 |
| 167.92 | 90 | 81.7 | 4.20 | 1.23 | 82.7 | 3.69 | 1.11 | 83.7 | 3.25 | 1.02 | 84.7 | 2.87 | .93 |
| 183.65 | 95 | 81.0 | 4.28 | 1.33 | 82.0 | 3.77 | 1.22 | 83.0 | 3.32 | 1.12 | 84.0 | 2.91 | 1.02 |
| 200.42 | 100 | 80.4 | 4.36 | 1.43 | 81.3 | 3.84 | 1.31 | 82.3 | 3.40 | 1.21 | 83.3 | 2.98 | 1.09 |
| 218.28 | 105 | 79.7 | 4.45 | 1.54 | 80.6 | 3.91 | 1.40 | 81.8 | 3.52 | 1.41 | 81.8 | 3.10 | 1.27 |
| 237.27 | 110 | 79.1 | 4.54 | 1.64 | 79.9 | 3.98 | 1.56 | 80.1 | 3.60 | 1.52 | 81.1 | 3.14 | 1.36 |
| 258.70 | 115 | 78.4 | 4.63 | 1.74 | 7.92 | 4.06 | .68 | 79.4 | 3.66 | 1.63 | 80.4 | 3.21 | 1.46 |
| 275.90 | 120 | 77.8 | 4.70 | 1.84 | 78.5 | 4.15 | 1.80 | | | | | | |

$$\text{Actual cubic feet per ton per minute} = \frac{\text{Theoretical cubic feet gas to produce one ton of refg.}}{\text{Volumetric efficiency}}$$

$$\text{I.hp. per ton} = \frac{\text{Plan}}{33{,}000 \times \text{Eff.}} \div \frac{\text{Plan}}{144 \times \text{Eff.} \times \text{Actual cubic feet per ton per minute}}$$

$$= \frac{P \times \text{Actual cubic feet per ton per minute}}{229}$$

TABLE 5

| Size of Compressor | Full Speed, R.p.m. | Cubic Feet Piston Displacement per Minute (Full Speed) | Cubic Feet Piston Displacement per Rev. C.F.R. | Per Cent Friction | |
|---|---|---|---|---|---|
| | | | | Steam drive | Belt drive |
| 1–6½ × 6 | 130 | 15.0 | .116 | 25 | 23 |
| 2–6½ × 6 | 130 | 30.0 | .231 | 24 | 22 |
| 7½ × 8 | 120 | 49.1 | .409 | 23 | 21 |
| 8½ × 10 | 110 | 72.3 | .657 | 22 | 20 |
| 9½ × 12 | 100 | 38.5 | .985 | 21 | 19 |
| 10½ × 15 | 95 | 142.8 | 1.50 | 20 | 18 |
| 12½ × 18 | 90 | 230.0 | 2.555 | 19 | 17 |
| 13½ × 20 | 85 | 281.5 | 3.312 | 18 | 16 |
| 15 × 24 | 80 | 392.7 | 4.91 | 17 | 16 |
| 16½ × 28 | 75 | 519.7 | 6.93 | 16 | 15 |
| 18 × 32 | 65 | 612.6 | 9.424 | 15 | 15 |
| 20½ × 36 | 60 | 825.2 | 13.75 | 15 | 15 |
| 22½ × 36 | 60 | 976.6 | 16.27 | 15 | 15 |
| 24 × 36 | 60 | 1131.0 | 18.85 | 15 | 15 |

FORMULAE

C.F. = Cu. ft. pist. displ. per ton per min. (From Table)
C.F.R. = Cu. ft. pist. displ. per rev. (From Table)
R.P.M. = Rev. per min. of compressor.
H.P. = Indicated amm. H.P. per ton of refrigeration. (From Table)
A.H.P. = Indicated ammonia H.P.
S.H.P. = Indicated Steam H.P.
B.H.P. = Belt H.P.
M.H.P. = H.P. of belted motor or eng.

1. Tons Ref. $= \dfrac{\text{C.F.R.} \times \text{R.P.M.}}{\text{C.F.}}$
2. A.H.P. = Tons ref. × H.P.
3. S.H.P. = A.H.P. × (1 + per cent "steam" friction).
4. B.H.P. = A.H.P. × (1 + per cent "belt" friction).
5. M.H.P. = B.H.P. × 110 per cent.

## TABLE 6

### Capacities of V. S. A. Enclosed Type Machines

| Cylinder | | | Piston Speed per Minute | | Displacement per Minute | | Capacity in Tons per 24 Hrs., 185-Lb. Condensing | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Number | Bore | Stroke | R p m. | Feet | Cubic inches | Cubic feet | 6 lb. −15° F | 9 lb. −10° F | 12 lb. −5° F | 16 lb. 0° F | | 19 lb. 5° F | 24 lb. 10° F | 28 lb. 15° F. | 33 lb. 20° F. |
| | | | | | | | Refg. | Refg. | Refg. | Ice | Refg. | Refg. | Refg. | Refg. | Refg. |
| 1 | 3 | 3 | 300 | 150 | 6,362 | 3.68 | .47 | .51 | .65 | .47 | .75 | .85 | 0.96 | 1.07 | 1.24 |
| 1 | 4 | 4 | 275 | 183 | 13,828 | 7.99 | .98 | 1.12 | 1.35 | 1.0 | 1.6 | 1.78 | 2.0 | 2.2 | 2.6 |
| 2 | 4 | 4 | 275 | 183 | 27,646 | 15.98 | 1.96 | 2.24 | 2.7 | 2.0 | 3.2 | 3.5 | 4.0 | 4.4 | 5.2 |
| 2 | 5 | 5 | 240 | 200 | 47,124 | 27.3 | 3.34 | 3.84 | 4.6 | 3.3 | 5.3 | 5.97 | 6.8 | 7.7 | 8.85 |
| 2 | 6 | 6 | 220 | 220 | 74,644 | 43.2 | 5.5 | 6.6 | 7.6 | 5.5 | 8.8 | 10.0 | 11.4 | 12.6 | 14.5 |
| 2 | 7 | 7 | 210 | 245 | 113,146 | 65.6 | 8.5 | 10.2 | 11.9 | 8.5 | 13.6 | 15.5 | 17.4 | 19.6 | 22.4 |
| 2 | 8 | 8 | 200 | 266 | 160,848 | 93.08 | 12.2 | 14.6 | 16.9 | 12.1 | 19.5 | 22.1 | 24.9 | 28.0 | 32.1 |
| 2 | 9 | 9 | 190 | 285 | 217,570 | 125.8 | 16.6 | 19.2 | 22.9 | 16.6 | 26.4 | 29.8 | 33.8 | 37.9 | 43.5 |
| 2 | 10 | 10 | 180 | 300 | 282,744 | 163.62 | 22.1 | 26.1 | 30.4 | 21.9 | 35.0 | 38.8 | 44.9 | 50.4 | 57.8 |
| 2 | 12 | 12 | 170 | 340 | 461,448 | 267.04 | 35.1 | 41.9 | 48.6 | 35.0 | 56.0 | 63.4 | 71.6 | 80.3 | 92.4 |

Multiply above capacities by the constants given below to obtain capacities at different condenser pressures given.

### Ammonia Suction Pressure

| Condenser Pressure Pounds, Gage | 6 Lb. | 9 Lb. | 12 Lb. | 16 Lb. | 19 Lb. | 24 Lb. | 28 Lb. | 33 Lb. |
|---|---|---|---|---|---|---|---|---|
| 200 | .975 | .969 | .975 | .977 | .978 | .977 | .976 | .977 |
| 168 | 1.029 | 1.028 | 1.028 | 1.028 | 1.02 | 1.027 | 1.026 | 1.022 |
| 153 | 1.046 | 1.049 | 1.05 | 1.05 | 1.043 | 1.05 | 1.042 | 1.042 |

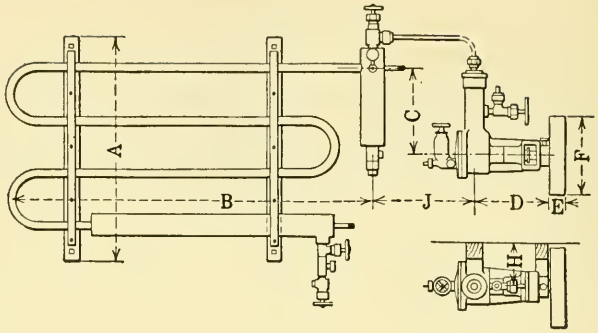


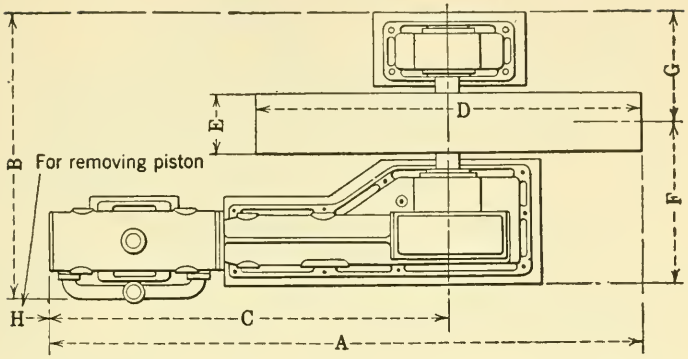

TABLE 7

| Enclosed Machine, with Oil Separator, Ammonia Condenser, Receiver and Connections | Number of Machine | | |
|---|---|---|---|
| | 00 | 0 | 1 |
| Heat units eliminated hourly, cooling 60 to 40 deg. F. | 4000 | 8000 | 16,000 |
| Ice-making capacity per 24 hrs.....tons | $\frac{1}{8}$ | $\frac{1}{4}$ | $\frac{1}{2}$ |
| Refrigerating capacity per 24 hrs.....tons | $\frac{1}{4}$ | $\frac{1}{2}$ | 1 |
| Actual horse power required | $\frac{3}{4}$ | $1\frac{1}{2}$ | $2\frac{3}{4}$ |
| Revolutions per minute | 250 | 250 | 200 |
| Shipping weight.....cwts. | 4 | 6 | 11 |
| Shipping measurement.....cu. ft. | 9 | 16 | 35 |

| Number | A | B | C | D | E | F | H | J |
|---|---|---|---|---|---|---|---|---|
| | Inches | Inches | Inches | Inches | Inches | Inches | Inches | Inches |
| 00 | 34 | 53 | 13 | $11\frac{1}{8}$ | $2\frac{1}{2}$ | 12 | $7\frac{1}{4}$ | 15 |
| 0 | 34 | 89 | 16 | $14\frac{3}{8}$ | 3 | 18 | 10 | 18 |
| 1 | 54 | 114 | 18 | 15 | 4 | 21 | 12 | 22 |

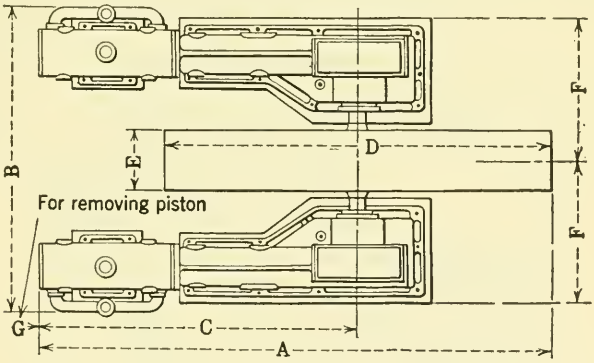
TABLE 8

ARCTIC ICE MACHINE CO. AMMONIA COMPRESSORS




| Cylinder Size | A | | B | | C | | D | | E | F | | G | | H | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Inches | Ft. | Ins. | Ft. | Ins. | Ft. | Ins. | Ft. | Ins. | Inches | Ft. | Ins. | Ft. | Ins. | Ft. | Ins. |
| 9 ×13½ | 13 | 8 | 5 | 11½ | 9 | 2 | 9 | 0 | 12 | 3 | 5¼ | 2 | 1½ | 3 | 1 |
| 10½×15¾ | 15 | 11½ | 7 | 2¼ | 10 | 11½ | 10 | 0 | 14 | 4 | 0⅝ | 2 | 7⅛ | 3 | 10 |
| 12 ×18 | 17 | 3⅞ | 7 | 10 7/16 | 11 | 3⅞ | 12 | 0 | 15 | 4 | 4⅜ | 2 | 9½ | 4 | 6 |
| 13½×20¼ | 19 | 11 9/16 | 8 | 11 7/16 | 12 | 5 9/16 | 12 | 0 | 22 | 4 | 11 | 3 | 1 | 4 | 9 |
| 15 ×22½ | 22 | 2 | 9 | 2 | 14 | 8 | 15 | 0 | 24 | 5 | 5 | 3 | 7½ | 5 | 0 |

Floor space required for single-cylinder, belted compressors

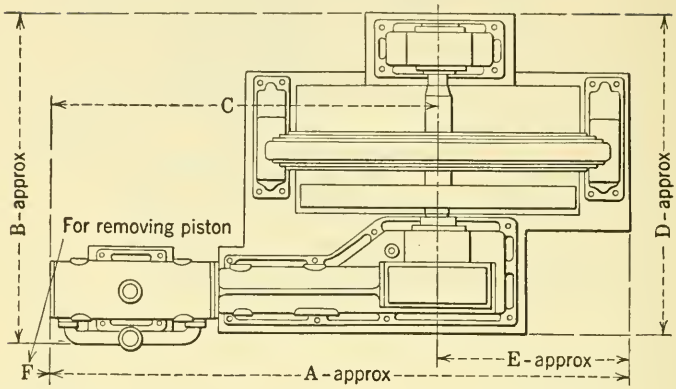



| Cylinder Size | A | | B | | C | | D | | E | F | | G | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Inches | Ft. | Ins. | Ft. | Ins. | Ft. | Ins. | Ft. | Ins. | Inches | Ft. | Ins. | Ft. | Ins. |
| 9 ×13½ | 13 | 8 | 8 | 8 | 9 | 2 | 9 | 0 | 22 | 3 | 11¼ | 3 | 1 |
| 10½×15¾ | 15 | 11½ | 9 | 2¼ | 10 | 11½ | 10 | 0 | 26 | 4 | 6⅝ | 3 | 10 |
| 12 ×18 | 17 | 3⅞ | 10 | 9⅞ | 11 | 3⅞ | 12 | 0 | 28 | 4 | 10⅞ | 4 | 6 |
| 13½×20¼ | 19 | 11 9/16 | 11 | 3⅞ | 12 | 5 9/16 | 15 | 0 | 28 | 5 | 2 | 4 | 9 |
| 15 ×22½ | 22 | 2 | 13 | 2 | 14 | 8 | 15 | 0 | 40 | 6 | 1 | 5 | 0 |

Floor space required for twin-cylinder, belted compressors

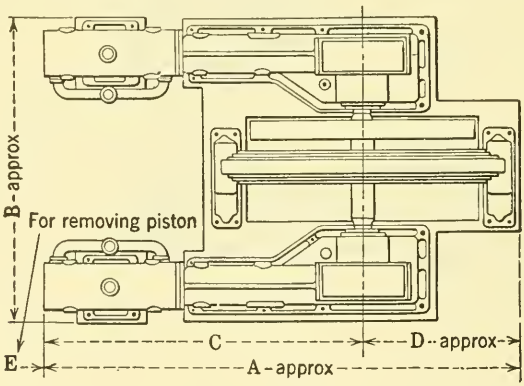
TABLE 9

ARCTIC ICE MACHINE AMMONIA COMPRESSORS




| Cylinder Size | Speed | A | | B | | C | | D | | E | | F | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Inches | | Ft. | Ins. | Ft. | Ins. | Ft. | Ins. | Ft. | Ins. | Ft. | Ins. | Ft. | Ins. |
| 9 ×13½ | 100–140 | 14 | 8 | 8 | 2 | 9 | 2 | 7 | 10 | 5 | 6 | 3 | 1 |
| 10½×15¾ | 100–140 | 16 | 7½ | 9 | 0 | 10 | 11½ | 8 | 6 | 5 | 8 | 3 | 10 |
| 12 ×18 | 90–130 | 18 | 0 | 11 | 0 | 11 | 3⅞ | 10 | 10 | 6 | 8⅛ | 4 | 6 |
| 13½×20¼ | 80–125 | 19 | 3 | 11 | 3 | 12 | 5 7/16 | 11 | 0 | 6 | 9 9/16 | 4 | 9 |
| 15 ×22½ | 75–120 | 22 | 5 | 13 | 0 | 14 | 8 | 12 | 4 | 7 | 9 | 5 | 0 |

Floor space required for Single-cylinder Compressors direct-connected to synchronous motors

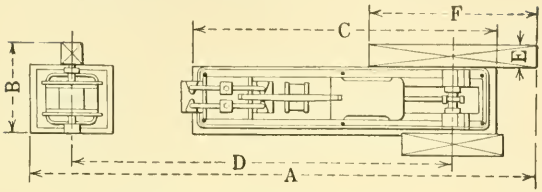



| Cylinder Size | A | | B | | C | | D | | E | |
|---|---|---|---|---|---|---|---|---|---|---|
| Inches | Ft. | Ins. | Ft. | Ins. | Ft. | Ins. | Ft. | Ins. | Ft. | Ins. |
| 9 ×13½ | 14 | 8 | 9 | 1½ | 9 | 2 | 5 | 6 | 3 | 1 |
| 10½×15¾ | 16 | 7½ | 9 | 11¼ | 10 | 11½ | 5 | 8 | 3 | 10 |
| 12 ×18 | 18 | 0 | 13 | 2 | 11 | 3⅞ | 6 | 8⅛ | 4 | 6 |
| 13½×20¼ | 19 | 3 | 13 | 6 | 12 | 5 7/16 | 6 | 9 9/16 | 4 | 9 |
| 15 ×22½ | 22 | 5 | 14 | 4 | 14 | 8 | 7 | 9 | 5 | 0 |

Floor space required for Twin-cylinder Compressors direct-connected to synchronous motors

## TABLE 10

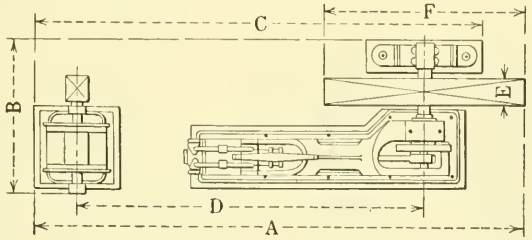
### American Carbonic $CO_2$ Compressors




### Horizontal Center Crank Type

Refrigerating Capacity, 3 to 10 Tons

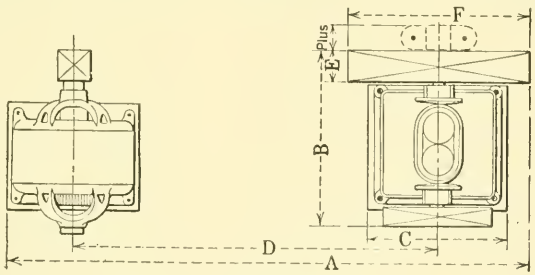
| Size in Tons, Refrigerating Capacity | A | | B | | C | | D | | E | F | | Overall Height | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Ft. | Ins. | Ft. | Ins. | Ft. | Ins. | Ft. | Ins. | Inches | Ft. | Ins. | Ft. | Ins. |
| 3 to 5 | 11 | 0 | 2 | 6 | 4 | 10½ | 8 | 0 | 6½ | 3 | 0 | 4 | 4 |
| 6 to 8 | 14 | 3 | 2 | 11 | 6 | 9¾ | 10 | 0 | 7½ | 5 | 0 | 5 | 6 |
| 8 to 10 | 14 | 6 | 2 | 11 | 6 | 10½ | 10 | 0 | 8½ | 5 | 0, | 5 | 6 |




### Horizontal Side Crank Type

Refrigerating Capacity, 10 to 150 Tons

| Size in Tons, Refrigerating Capacity | A | | B | | C | | D | | E | F | | Overall Height | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Ft. | Ins. | Ft. | Ins. | Ft. | Ins. | Ft. | Ins. | Inches | Ft. | Ins. | Ft. | Ins. |
| 10 to 15 | 17 | 0 | 4 | 8½ | 15 | 4 | 12 | 0 | 12 | 6 | 10 | 7 | 5 |
| 18 to 25 | 19 | 0 | 5 | 3 | 17 | 6 | 14 | 0 | 12 | 7 | 0 | 7 | 7 |
| 30 to 40 | 21 | 10 | 5 | 9¾ | 20 | 1 | 16 | 0 | 17½ | 7 | 11 | 7 | 11 |
| 45 to 65 | 30 | 0 | 7 | 1 | 26 | 10 | 21 | 0 | 20 | 12 | 0 | 9 | 9 |
| 70 to 100 | 32 | 0 | 9 | 0 | 29 | 9 | 22 | 0 | 26 | 12 | 0 | 9 | 9 |
| 125 to 150 | 36 | 6 | 10 | 0 | 34 | 6 | 26 | 0 | 30 | 14 | 0 | 9 | 9 |



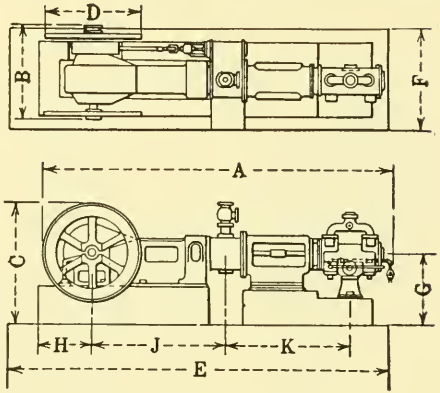

### Vertical Type

Refrigerating Capacity, 15 to 180 Tons

| Size in Tons, Refrigerating Capacity | A | | B | | C | | D | | E | F | | Overall Height | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Ft. | Ins. | Ft. | Ins. | Ft. | Ins. | Ft. | Ins. | Inches | Ft. | Ins. | Ft. | Ins. |
| 15 to 25 | 14 | 6 | 5 | 0 | 3 | 10 | 10 | 0 | 11 | 5 | 0 | 6 | 8 |
| 25 to 35 | 15 | 6 | 5 | 0 | 3 | 10 | 10 | 0 | 11 | 5 | 0 | 6 | 8 |
| 40 to 50 | 15 | 0 | 5 Plus 12 | 0 | 3 | 6 | 10 | 0 | 16 | 5 | 0 | 7 | 0 |
| 60 to 100 | 17 | 6 | 7 Plus 18 | 0 | 3 | 6 | 12 | 0 | 18 | 5 | 0 | 7 | 0 |
| 120 to 180 | 20 | 0 | 9 Plus 20 | 6 | 3 | 6 | 14 | 0 | 28 | 5 | 0 | 7 | 0 |

TABLE 11

WORTHINGTON PUMP CO. AMMONIA COMPRESSOR




Steam-driven Compressors
(Layout Dimensions)

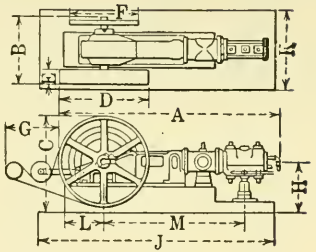
| Size | A | | B | | C | | D | | E | | F | | G | | H | | J | | K | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Ft. | In. | Ft. | In. | Ft. | In. | Ft. | In. | Ft. | In. | Ft. | In. | Ft. | In. | Ft. | In. | Ft. | In. | Ft. | In. |
| 8 × 6 × 9 p | 11 | 0 | 2 | 9 | 3 | 10 | 3 | 0 * | 10 | 0 | 3 | 4 | 2 | 0½ | 0 | 18 | 3 | 11 | 4 | 4 |
| 9 × 7 ×10 p | 12 | 11 | 3 | 3 | 4 | 4 | 3 | 6 | 13 | 0 | 4 | 2 | 2 | 6½ | 0 | 24 | 4 | 10½ | 5 | 0½ |
| 10 × 8 ×12 p | 14 | 3 | 3 | 7 | 4 | 7 | 4 | 0 | 14 | 0 | 4 | 6 | 2 | 6½ | 2 | 3 | 5 | 6 | 5 | 4 |
| 10½ × 8 ×12 u | 15 | 6 | 3 | 7 | 4 | 6½ | 4 | 0 | 16 | 0 | 4 | 6 | 2 | 6½ | 2 | 3 | 6 | 3½ | 5 | 11 |
| 12 × 9 ×14 p | 16 | 0 | 4 | 6 | 5 | 0 | 4 | 6 | 17 | 0 | 4 | 8 | 2 | 8½ | 2 | 6 | 6 | 1¾ | 5 | 9¾ |
| 12½ × 9 ×14 u | 17 | 9 | 4 | 0¾ | 4 | 11½ | 4 | 6 | 19 | 0 | 4 | 8 | 2 | 8½ | 2 | 6 | 7 | 2½ | 6 | 7¾ |
| 14 ×10 ×16 p | 18 | 3 | 4 | 8 | 5 | 4 | 5 | 0 | 19 | 0 | 5 | 0 | 2 | 10 | 2 | 6 | 7 | 0 | 6 | 5¾ |
| 15 ×10 ×16 u | 19 | 6 | 4 | 7 | 5 | 5½ | 5 | 0 | 20 | 6 | 5 | 0 | 3 | 1 | 2 | 6 | 8 | 0½ | 7 | 0¼ |
| 16½ ×11½ ×18 n | 23 | 6 | 8 | 0 | 6 | 6 | 6 | 6 | 24 | 0 | 5 | 6 | 3 | 0 | 3 | 0 | 9 | 2 | 7 | 11 |
| 18½ ×13 ×21 n | 25 | 6 | 8 | 9 | 6 | 6 | 7 | 0 | 26 | 0 | 6 | 0 | 3 | 0 | 3 | 3 | 10 | 2 | 8 | 6 |
| 20½ ×14 ×24 n | 27 | 6 | 10 | 6 | 7 | 6 | 8 | 0 | 28 | 0 | 7 | 0 | 3 | 6 | 3 | 6 | 11 | 8 | 9 | 3 |
| 23½ ×16½ ×24 n | 28 | 6 | 11 | 0 | 8 | 0 | 8 | 0 | 30 | 0 | 7 | 0 | 3 | 9 | 3 | 8 | 12 | 0 | 9 | 6 |

*A 36 in. ×8½ in. pulley only on 8 ×6 ×9 compressor.

p Compressors fitted with piston valve steam cylinders as illustrated.
u Compressors fitted with uniflow piston valve steam cylinders (not illustrated).
n Compressors fitted with uniflow poppet valve steam cylinders (not illustrated).
Foot under cylinders not provided for 8 ×6 ×9 and 9 ×7 ×10 compressors.

# TABLE 12

## Worthington Pump Co. Ammonia Compressor




### Single-belted compressors

(Layout Dimensions)

| Compressor Size | A | | B | | C | | D | | E | F | | G | | H | | J | | K | | L | | M | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Ft. | In. | Ft. | In. | Ft. | In. | Ft. | In. | Inches | Ft. | In. | Ft. | In. | Ft. | In. | Ft. | In. | Ft. | In. | Ft. | In. | Ft. | In. |
| 6 × 9 | 7 | 6 | 2 | 4½ | 4 | 1 | 3 | 6 | 8½ | ..... | | 2 | 6 | 2 | 4 | 10 | 3 | 3 | 4 | | 18 | 5 | 5 |
| 7 ×10 | 10 | 10 | 3 | 3⅝ | 4 | 9½ | 4 | 6 | 8½ | 3 | 6 | 3 | 3 | 2 | 6½ | 11 | 0 | 4 | 2 | | 24 | 7 | 5 |
| 8 ×12 | 11 | 10 | 3 | 10 | 5 | 5 | 5 | 0 | 10½ | 4 | 0 | 3 | 6 | 2 | 10¾ | 13 | 6 | 4 | 6 | 2 | 3 | 8 | 0¾ |
| 9 ×14 | 13 | 2 | 4 | 6⅛ | 6 | 0 | 5 | 6 | 12½ | 4 | 6 | 4 | 0 | 3 | 2¾ | 15 | 0 | 4 | 8 | 2 | 6 | 8 | 10 |
| 10 ×16 | 14 | 4 | 5 | 5 | 7 | 1 | 6 | 6 | 15½ | 5 | 0 | 3 | 9 | 3 | 9½ | 16 | 0 | 5 | 0 | 2 | 6 | 9 | 9¾ |

### Electrically-driven Type

(Not Illustrated)

| | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 7 ×10 | 11 | 7 | 7 | 0 | 4 | 9 | 4 | 6 | ..... | ..... | | ..... | | 2 | 0 | 11 | 0 | 10 | 0 | 3 | 0 | 7 | 5 |
| 8 ×12 | 12 | 4 | 7 | 6 | 5 | 4 | 5 | 0 | ..... | ..... | | ..... | | 2 | 4 | 12 | 0 | 11 | 6 | 3 | 6 | 8 | 0¾ |
| 10 ×14 | 12 | 8 | 8 | 0 | 5 | 8 | 5 | 4 | ..... | ..... | | ..... | | 2 | 6 | 13 | 3 | 12 | 6 | 3 | 6 | 8 | 10¼ |
| 11½ ×16 | 15 | 8 | 9 | 0 | 6 | 4 | 6 | 0 | ..... | ..... | | ..... | | 2 | 10 | 15 | 6 | 13 | 0 | 4 | 0 | 9 | 9¾ |
| 13 ×18* | 17 | 0 | 10 | 6 | 7 | 0 | 6 | 6 | ..... | ..... | | ..... | | 3 | 0 | 17 | 6 | 14 | 0 | 4 | 6 | 9 | 6 |
| 14½ ×21* | 18 | 6 | 11 | 6 | 7 | 0 | 7 | 0 | ..... | ..... | | ..... | | 3 | 3 | 18 | 6 | 14 | 6 | 4 | 9 | 10 | 0 |
| 16½ ×24* | 20 | 0 | 12 | 0 | 7 | 3 | 7 | 0 | ..... | ..... | | ..... | | 3 | 3 | 20 | 0 | 14 | 9 | 5 | 0 | 10 | 10 |

Dimensions "B" and "K" are for electrically-driven machines giving maximum width with outboard bearing motor. If a Flywheel type or Overhung motor is used, width is about the same as for a belted machine. Motor prints are necessary to secure exact dimensions.

* Do not have "Doubleseal" Stuffing Box, 18-in. stroke and over.

## TABLE 13

### Ingersoll-Rand Ammonia Compressor

### "Imperial" Type XPV-2-A Two-stage Steam-driven Ammonia Compressors

*Sea Level Operation*

Duplex Steam Cylinders—Compound Steam Cylinders Furnished, Sizes Depending upon Steam Conditions

"UNIVERSAL MACHINES"

| Cylinder Dimensions, Inches | | | | Pressure Conditions, Pounds per Square Inch, Gage | | | | Overall Dimensions, Feet and Inches | | | Approximate Shipping Weight, Pounds | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Diameters | | | Stroke | Suction—20 Discharge—185 | | Suction—5 Discharge—185 | | Length | Width | Height | Domestic | Export |
| Duplex steam cylinders | Ammonia cylinders: Low pressure | Ammonia cylinders: High pressure | | Tons refrigeration | I. H. P. steam cylinders | Tons refrigeration | I. H. P. steam cylinders | | | | | |
| 7* | 7½ | 5¼ | 10 | 30 | 48 | 17 | 39 | 9–0 | 5–1 | 6–4 | 8,000 | 8,800 |
| 8* | 8½ | 6 | 12 | 42 | 66 | 24 | 54 | 9–10 | 5–4 | 6–5 | 8,700 | 9,600 |
| 9* | 9½ | 6¾ | 12 | 53 | 82 | 31 | 67 | 10–11 | 6–0 | 6–9 | 11,300 | 12,500 |
| 10 | 12 | 8½ | 14 | 88 | 129 | 51 | 105 | 12–6 | 7–8 | 7–1 | 19,900 | 22,000 |
| 12 | 13½ | 9½ | 16 | 114 | 166 | 66 | 135 | 14–0 | 8–9 | 6–8 | 26,000 | 28,500 |
| 14 | 15 | 10½ | 16 | 142 | 207 | 83 | 168 | 14–6 | 9–2 | 6–5 | 33,000 | 37,000 |
| "ICE MACHINES" ONLY—20 Lb. per Square Inch Gage Suction and 185 Lb. per Square Inch Gage Discharge Pressures | | | | | | | | | | | | |
| 15 | 16½ | 11½ | 20 | 182 | 258 | ..... | ..... | 17–10 | 10–3 | 6–6 | 43,000 | 47,500 |
| 16 | 18 | 12½ | 20 | 218 | 310 | ..... | ..... | 18–5 | 11–0 | 6–11 | 50,300 | 56,000 |
| 19 | 23 | 16 | 24 | 357 | 496 | ..... | ..... | 23–8 | 15–6 | 9–6 | 84,000 | 92,500 |
| 22 | 26 | 18 | 30 | 506 | 702 | ..... | ..... | 28–4 | 17–0 | 10–0 | 123,000 | 136,000 |
| "FREEZER MACHINES" ONLY—5 Lb. per Square Inch Gage Suction and 185 Lb. per Square Inch Discharge Pressures | | | | | | | | | | | | |
| 15† | 17½ | 11 | 20 | ..... | ..... | 117 | 236 | 17–10 | 10–3 | 6–6 | 44,000 | 48,500 |
| 16† | 20 | 12 | 20 | ..... | ..... | 151 | 303 | ..... | ..... | 6–11 | 52,000 | 58,000 |
| 19† | 25 | 16 | 24 | ..... | ..... | 248 | 485 | 23–8 | 15–6 | 9–6 | 86,000 | 94,000 |
| 22† | 28 | 18 | 30 | ..... | ..... | 348 | 680 | 28–4 | 17–0 | 10–0 | 125,000 | 137,500 |

* Steam Cylinders have plain piston Steam Valves—Steam Governor is Variable Speed Type.
† Low-Pressure Ammonia Cylinder has Clearance Pockets.

TABLE 14

DETAILS OF STANDARD CARBONIC COMPRESSORS

| Stroke | Tons | Bore | Diameter of Piston Rod, Inches | R.p.m. | Total Horse Power | Horse Power per Ton | Suction Displacement | | | Flywheel | | Total Weight, Pounds |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | Cubic inches per minute | Cubic feet per minute | Cubic inches per min.-ton | Weight, pounds | Diameter, inches | |
| 6 | 3 | 2¼ | 15/16 | 120 | 7½ | 2.5 | 5,220 | 3.02 | 1740 | 1,000 | 36 | 2,100 |
| 6 | 4 | 2¼ | 15/16 | 135 | 10 | 2.5 | 5,870 | 3.40 | 1400 | 1,000 | 36 | 2,100 |
| 9 | 5 | 2½ | 1⅛ | 95 | 10 | 2.0 | 7,550 | 4.37 | 1510 | 1,700 | 60 | 3,800 |
| 9 | 6 | 2¾ | 1⅛ | 110 | 15 | 2.5 | 10,750 | 6.23 | 1800 | 1,700 | 60 | 3,800 |
| 10 | 7 | 2⅞ | 1⅛ | 110 | 20 | 2.8 | 13,200 | 7.65 | 1885 | 2,200 | 60 | 4,300 |
| 10 | 8 | 2⅞ | 1⅛ | 110 | 20 | 2.5 | 13,200 | 7.65 | 1650 | 2,200 | 60 | 4,300 |
| 11½ | 10 | 3 | 1¼ | 90 | 20 | 2.0 | 13,300 | 7.7 | 1330 | 2,600 | 82 | 6,200 |
| 11½ | 12 | 3¼ | 1¼ | 90 | 25 | 2.07 | 15,900 | 9.22 | 1320 | 2,600 | 82 | 6,200 |
| 14 | 15 | 3½ | 1¾ | 90 | 25 | 1.66 | 21,300 | 12.32 | 1420 | 3,000 | 82 | 9,500 |
| 14 | 18 | 3⅝ | 1¾ | 85 | 30 | 1.66 | 21,700 | 12.58 | 1200 | 4,300 | 84 | 10,500 |
| 14 | 20 | 3¾ | 1¾ | 95 | 35 | 1.74 | 26,400 | 15.3 | 1320 | 4,500 | 84 | 11,000 |
| 14 | 22 | 3⅞ | 1¾ | 95 | 40 | 1.8 | 28,300 | 16.38 | 1280 | 4,500 | 84 | 11,000 |
| 16 | 25 | 4¼ | 2 | 80 | 40 | 1.6 | 32,300 | 18.69 | 1290 | 6,000 | 96 | 14,500 |
| 16 | 30 | 4⅜ | 2 | 90 | 50 | 1.66 | 38,700 | 22.4 | 1290 | 6,500 | 96 | 15,000 |
| 20 | 40 | 5 | 2⅜ | 70 | 60 | 1.5 | 48,700 | 28.21 | 1220 | 8,000 | 108 | 20,000 |
| 20 | 50 | 5⅛ | 2⅜ | 70 | 75 | 1.5 | 51,500 | 29.8 | 1030 | 9,000 | 108 | 21,000 |
| 20 | 60 | 5¾ | 2⅜ | 75 | 90 | 1.5 | 66,500 | 38.5 | 1110 | 10,000 | 144 | 22,000 |
| 24 | 70 | 6¼ | 3 | 65 | 100 | 1.42 | 88,000 | 50.9 | 1250 | 11,500 | 144 | 25,000 |
| 24 | 80 | 6½ | 3 | 65 | 110 | 1.38 | 96,000 | 55.6 | 1206 | 14,000 | 168 | 27,500 |
| 24 | 90 | 6¾ | 3 | 65 | 120 | 1.34 | 101,000 | 58.5 | 1120 | 15,000 | 168 | 28,500 |
| 24 | 100 | 7 | 3 | 65 | 135 | 1.35 | 109,000 | 63.1 | 1090 | 16,000 | 168 | 29,500 |
| 28 | 120 | 7¾ | 3¾ | 60 | 150 | 1.26 | 120,000 | 69.5 | 1000 | 18,000 | 192 | 48,000 |
| 28 | 130 | 8 | 3¾ | 60 | 200 | 1.52 | 128,000 | 74.4 | 990 | 20,000 | 216 | 50,000 |
| 28 | 150 | 8½ | 3¾ | 60 | 275 | 1.82 | 147,000 | 85.2 | 970 | 25,000 | 216 | 55,000 |

## CHAPTER III

## THE ABSORPTION REFRIGERATING MACHINE

**History and Development.**—The absorption machine was one of the first forms of refrigerating devices. It is based on an entirely different principle from that of the compression machine, namely that of driving a gas out of solution by the direct application of heat and (after evaporation in the refrigerating coils) of absorbing this gas into solution again in order to complete the cycle. The first mention of such a method is that by Professor Leslie who, about 1810, described a laboratory method of making ice by freezing water in a saucer placed in a receiver under a vacuum—the water vapor as it boiled off under the low pressure maintained in the receiver being absorbed by sulphuric acid in another saucer. Much later, in 1878, Windhauser made a complete cycle, using heat to drive the water out of solution and to obtain a concentrated solution again. Edmund Carré[1] in 1850 developed a small sulphuric acid refrigerating machine of the intermittent type, and the ammonia absorption machine of the present day is the old Carré perfected.

**The Modern Design.**—In the ammonia absorption machine, the *generator* (Fig. 55) contains a concentrated solution of ammonia and water and heat is applied[2] by means of steam coils submerged in the solution. The gas driven off is later condensed in an ordinary ammonia condenser and on evaporation in the refrigerating coils is reabsorbed in the *absorber* (Figs. 56 and 57). The absorption medium is the weaker aqueous solution of ammonia from the generator, which has to be cooled to some temperature in the neighborhood of 100 deg. F. or lower in order to work with concentrations required in the present practice. In 1870 Mart devised the *economizer* or *exchanger* (a modern design of which is shown in Fig. 58) whereby this weaker solution was cooled by a counter current flow exchange of heat with the stronger and colder solution from the absorber. This operation permitted the stronger

[1] Described by Dr. John Hopkinson in 1882 before the Society of Arts. Such an ice plant made about 12 tons of ice per 24 hours in 650-lb. blocks.

[2] In the household refrigerating machine, gas or electric resistance coils provide the necessary heating.

solution to return to the generator at as high a temperature and the weaker solution to arrive at the absorber at as low a temperature as possible. The absorption machine devised by Pontifex and Wood in 1876 included the *analyzer* containing trays invented by Reece to separate from the vapors leaving the generator some of the 5 to 15 per cent of water vapor which is present in the ammonia vapor as well as the particles of liquid (priming) which are carried on by the action of boiling in the generator. The analyzer is really a part of the generator, but it is omitted in some designs. At present a *rectifier* (or dehydrator) is also used in order to condense out as much of the water vapor as can safely be attempted by cooling these gases in a sort of water-cooled pre-cooler or preliminary condenser where only part of the superheat in the ammonia is taken away. A liquid pump, the only moving part of the

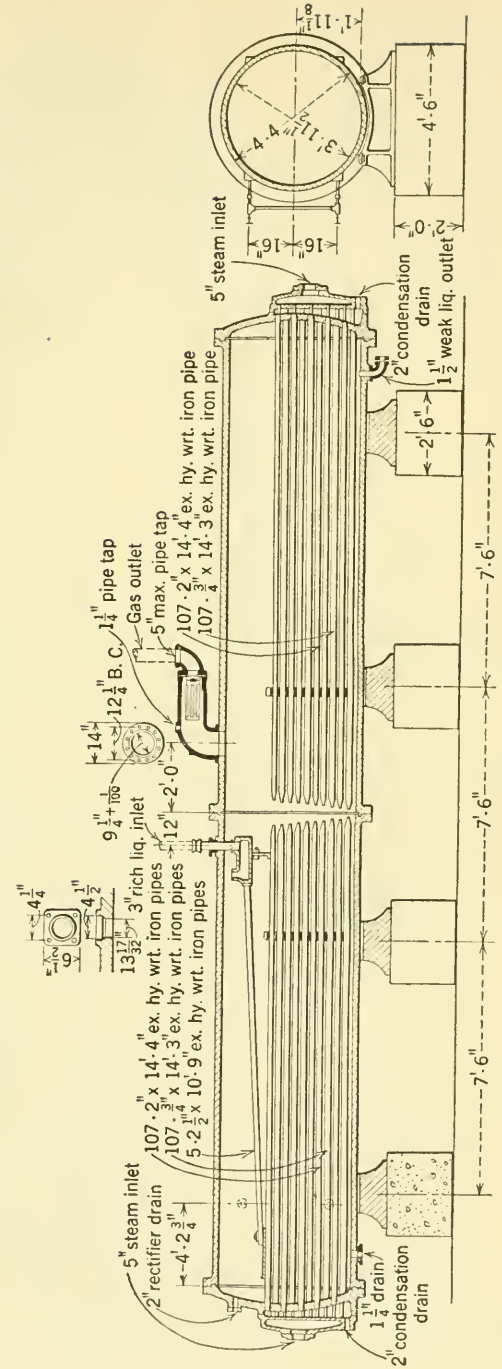



Fig. 55.—The Generator.

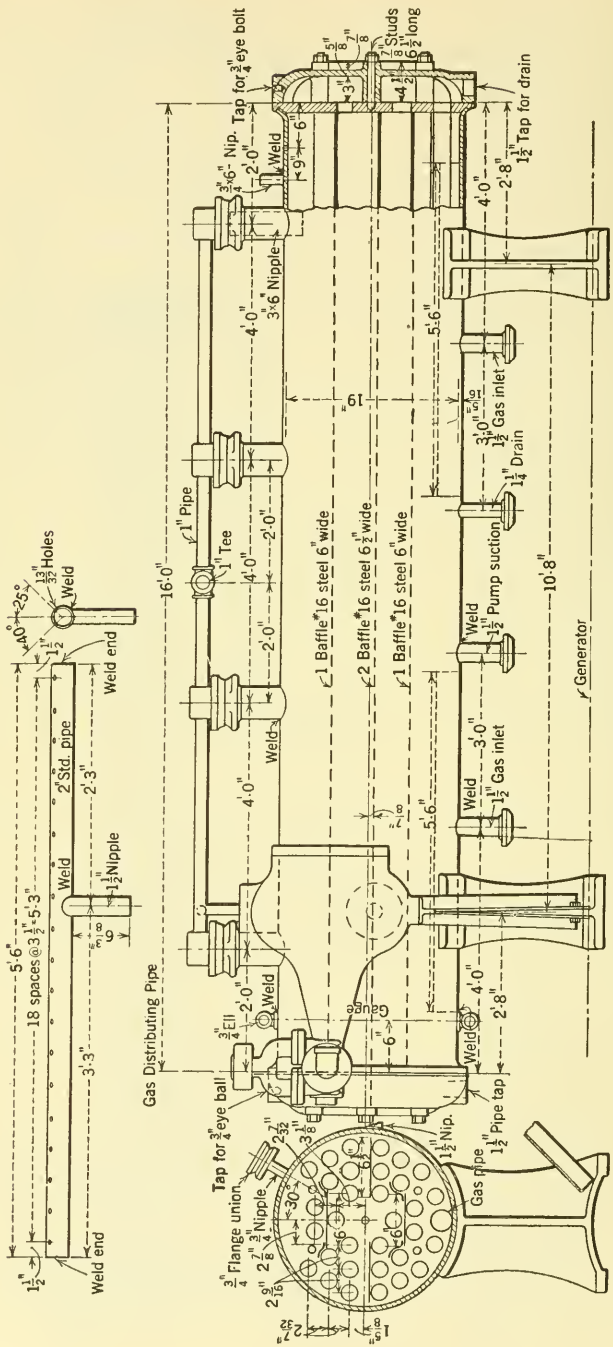



FIG. 56.—The Absorber.

machine, returns the strong aqua to the generator, the complete cycle being shown in Fig. 59.

The absorption machine has never been very popular in the United States for general refrigeration (but much more so than in Great Britain) although in special work it has certain advantages over the compression type: for example, those plants having a large amount of exhaust steam for which no other use can be found frequently can use the absorption in preference to the compression machine. This appears to be the case in oil refineries and (to some extent) in packing plants, where usually exhaust steam is plentifully available. The use of high-pressure steam in the coils of the generator is now no longer advocated although many such plants were installed up to about 1915. The use of the absorption machine for the production of low temperatures is now no longer necessary, although for a long time low temperature work was considered the field for the absorption machine only. In this respect it should be noted that the absorber liquefies and then absorbs the gas from the evaporating coils and is in no manner affected by the pressure exerted by the gas (and in consequence the specific volume of this gas) provided the pipe from the evaporator to the absorber is large enough to accommodate the increased volume at these low pressures, and that the absorber can spray the entering gas properly with the weak solution. The specific volume of the ammonia therefore is not an important factor in the absorption machine, though it is extremely important in the compression type, both because of the piston displacement required and also because of the decreased volumetric efficiency occasioned by the cylinder walls and ports. As stage ammonia compression has been developed so that it can take low temperature refrigeration with economy the use of the absorption machine for low temperature work is no longer necessary, with the result that, as mentioned, the best field for this machine is where exhaust steam would otherwise be thrown away.

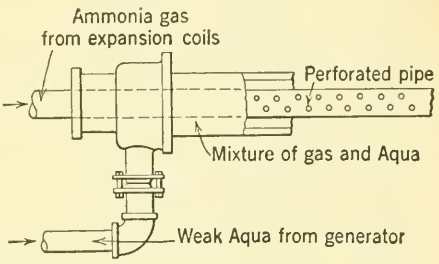



Fig. 57.—Weak Aqua and Gas Mixer.

The principle of the operation of the absorption machine[3] is the ability of water to absorb large quantities of ammonia—as much at

[3] The values of the $p$, $t$, $x$ relations of aqua ammonia are given by Thomas Wilson in Bulletin No. 146 of the Engineering Experiment Station of the University of

times as 1000 volumes—depending in amount on the temperature and the pressure of the solution (Table 18). For example a concentration of 42.5 per cent is obtained in the absorber under a pressure of 35 lb. abs. (20 lb. gage) and 80 deg. F., whereas at a temperature of 200 degrees and a pressure of 165 lb. abs. (150 lb. gage) the maximum concentration that can be held in solution is 35.5 per cent, and the difference, 7.0 per cent, will represent the amount to be boiled out of solution in the generator.

**Action of the Generator.**—The action in the generator is not as clearly understood as it might be, but the following explanation is at

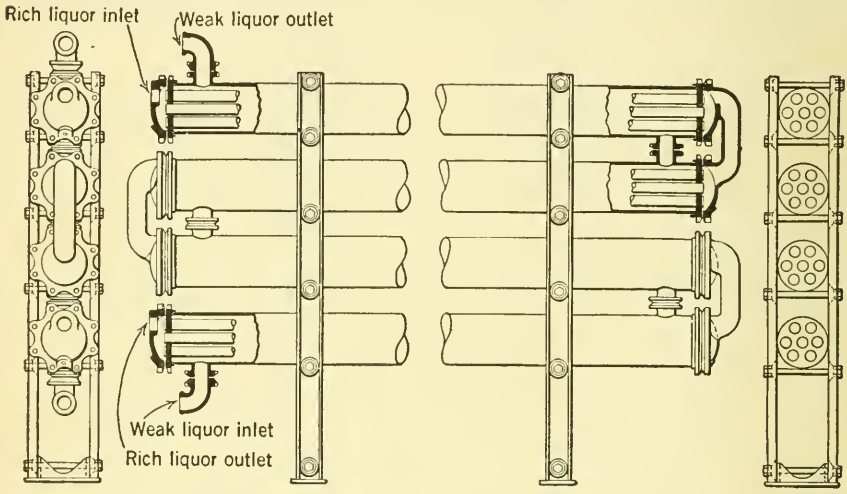



FIG. 58.—The Exchanger.

least reasonable. The heat supplied by the steam in the steam coils can be considered to cause the following action:

1. To break up the bond of association between the ammonia and the water.

2. To cool the resulting anhydrous ammonia to the temperature corresponding to saturation for the pressure at hand.

Illinois. The equation connecting these variables is given by G. A. Goodenough as,

$$\frac{\theta}{T} = \frac{1}{1 + 0.70356(1 - x^2)}$$

$$z = \sqrt{x + 0.05}(1.347 - 2.9x + 1.77x^2)$$

$\theta$ = the temperature of saturated ammonia vapor, deg. abs.
$T$ = the temperature of ammonia solution, deg. abs
$x$ = mol concentration of ammonia.

## TABLE 15

### Heat Removed in Absorber for Different Values of Strong and Weak Aqua

| Per Cent of Weak Aqua | Temperature and Concentration of Strong Aqua | | | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 25 per cent | | | | | | | | | 30 per cent | | | | | | | | | 35 per cent | | | | | | | | | 40 per cent | | | | | |
| | 60 deg. F. | | | 80 deg. F. | | | 100 deg. F. | | | 60 deg. F. | | | 80 deg. F. | | | 100 deg. F. | | | 60 deg. F. | | | 80 deg. F. | | | 100 deg. F. | | | 60 deg. F. | | | 80 deg. F. | | | 100 deg. F. | | |
| | Temperature Weak Aqua | | | | | | | | | Temperature Weak Aqua | | | | | | | | | Temperature Weak Aqua | | | | | | | | | Temperature Weak Aqua | | | | | | | | |
| | 80 | 100 | 120 | 100 | 120 | 140 | 120 | 140 | 160 | 80 | 100 | 120 | 100 | 120 | 140 | 120 | 140 | 160 | 80 | 100 | 120 | 100 | 120 | 140 | 120 | 140 | 160 | 80 | 100 | 120 | 100 | 120 | 140 | 120 | 140 | 160 |
| 10 G | 6.0 | 6.0 | 6.0 | 6.0 | 6.0 | 6.0 | 6.0 | 6.0 | 6.0 | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| H | 853 | 954 | 1056 | 838 | 939 | 1041 | 821 | 924 | 1026 | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| 15 G | 8½ | 8½ | 8½ | 8½ | 8½ | 8½ | 8½ | 8½ | 8½ | 5⅔ | 5⅔ | 5⅔ | 5⅔ | 5⅔ | 5⅔ | 5⅔ | 5⅔ | 5⅔ | | | | | | | | | | | | | | | | | | |
| H | 892 | 1046 | 1200 | 877 | 1032 | 1186 | 862 | 1016 | 1173 | 828 | 923 | 1019 | 812 | 908 | 1004 | 795 | 891 | 989 | | | | | | | | | | | | | | | | | | |
| 20 G | 16 | 16 | 16 | 16 | 16 | 16 | 16 | 16 | 16 | 8 | 8 | 8 | 8 | 8 | 8 | 8 | 8 | 8 | 5⅓ | 5⅓ | 5⅓ | 5⅓ | 5⅓ | 5⅓ | 5⅓ | 5⅓ | 5⅓ | | | | | | | | | | |
| H | 1035 | 1344 | 1656 | 1021 | 1332 | 1644 | 1006 | 1318 | 1636 | 862 | 1007 | 1153 | 848 | 993 | 1139 | 831 | 976 | 1125 | 799 | 889 | 979 | 783 | 873 | 963 | 766 | 856 | 948 | | | | | | | | | | |
| 25 G | .... | .... | .... | .... | .... | .... | .... | .... | .... | 15 | 15 | 15 | 15 | 15 | 15 | 15 | 15 | 15 | 7½ | 7½ | 7½ | 7½ | 7½ | 7½ | 7½ | 7½ | 7½ | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 | 5 |
| H | .... | .... | .... | .... | .... | .... | .... | .... | .... | 995 | 1285 | 1579 | 980 | 1274 | 1568 | 964 | 1258 | 1559 | 830 | 966 | 1102 | 814 | 950 | 1087 | 798 | 934 | 1074 | 769 | 852 | 936 | 752 | 836 | 920 | 735 | 819 | 905 |
| 30 G | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 14 | 14 | 14 | 14 | 14 | 14 | 14 | 14 | 14 | 7 | 7 | 7 | 7 | 7 | 7 | 7 | 7 | 7 |
| H | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 952 | 1224 | 1499 | 937 | 1211 | 1487 | 921 | 1197 | 1480 | 796 | 922 | 1047 | 779 | 906 | 1033 | 762 | 890 | 1020 |
| 35 G | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 13 | 13 | 13 | 13 | 13 | 13 | 13 | 13 | 13 |
| H | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 907 | 1160 | 1416 | 891 | 1147 | 1404 | 876 | 1132 | 1397 |

G = lb. of Strong Aqua per 1 lb. of Ammonia absorbed.
H = B.t.u. absorbed by the cooling water per 1 lb. of Ammonia entering solution.

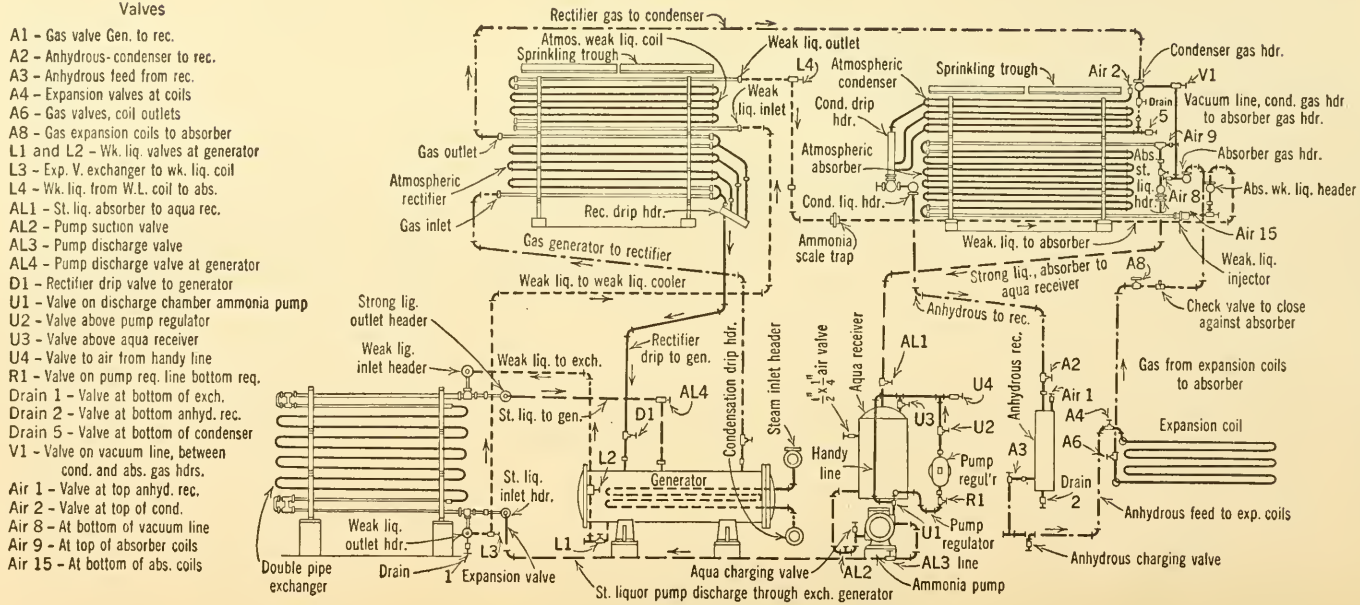



FIG. 59.—The Absorption Machine Complete.

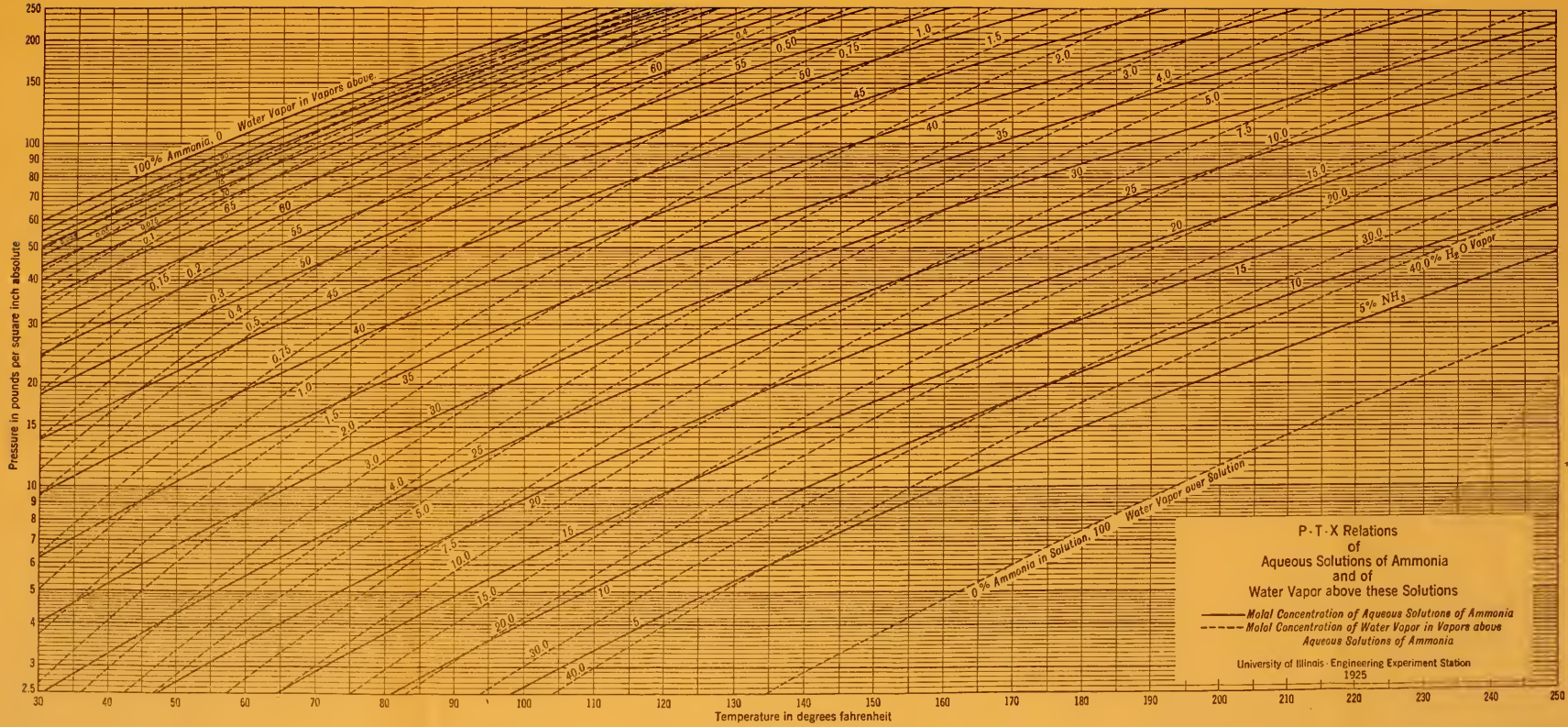

Pressure in pounds per square inch absolute
Temperature in degrees fahrenheit
100% Ammonia, 0 — Water Vapor in Vapors above
0% Ammonia in Solution, 100 — Water Vapor over Solution
40.0% $H_2O$ Vapor
5% $NH_3$
P·T·X Relations
of
Aqueous Solutions of Ammonia
and of
Water Vapor above these Solutions
Molal Concentration of Aqueous Solutions of Ammonia
Molal Concentration of Water Vapor in Vapors above Aqueous Solutions of Ammonia
University of Illinois · Engineering Experiment Station
1925

FIG. 60.—Chart of Properties of Aqueous Solutions of Ammonia.

*To face page 89.*

3. To vaporize the liquid ammonia.

4. To heat the entering strong aqua from the absorber and the exchanger to the temperature of the generator, and, finally,

5. To evaporate a certain amount of water vapor which passes off into the rectifier with the vapor boiled off. The heat of solution is given by the formula [4]

$$q = 345(1 - x) - 400x^2$$

where $x$ is the average concentration of the solution. The following problem will bring out the method of calculation.

**Problem.**—A strong aqueous solution enters the generator at 160 deg. F. and a concentration of 35 per cent, and the temperature of the solution in the generator is 215 degrees. The weak aqua leaving the generator has a concentration of 30 per cent and the pressure in the generator is 145 lb. per square inch. Water vapor to the amount of 7.4 per cent is boiled off with the ammonia.

The amount of strong aqua necessary to be circulated in order to boil off one pound of ammonia is given by the relation,

$$nx_1 - (n - 1)x_2 = 1$$

where $x_1$ = the concentration of the strong
$x_2$ = the concentration of the weak

and therefore

$$n = \frac{1 - x_2}{x_1 - x_2} = \frac{1.0 - 0.30}{0.35 - 0.30} = 14.0 \text{ lb.}$$

The heat supplied by the steam coils in the generator then becomes:

*a.* Heat of disassociation

$$q = 345(1 - x) - 400x^2 = 345 \times 0.675 - 400 \times 0.325^2 = 190.8 \text{ B.t.u.}$$
per 1.0 lb. ammonia boiled out.

*b.* Heat necessary to cool the liquid ammonia from 215 to 76.8 deg.

$= -305 + 128.4 = -176.6$ B.t.u.

*c.* Heat necessary to vaporize the ammonia (at 145 lb.)

$= 501.9$ B.t.u.

*d.* Heat necessary to superheat this gas to 215 degrees

$= 88.3$ B.t.u.

*e.* Heat required to raise the temperature of the entering strong aqua[5]

$= 1.132 \times (215 - 160) \times 14.0 = 871$ B.t.u.

*f.* Finally the heat necessary to boil off 0.074 lb. of water vapor at 215 deg. F. is

$= 0.074 \times 969.9 = 71.8$ B.t.u.

[4] H. Mollier, V. D. I., 1907. Goodenough, Principles of Thermodynamics.
[5] Ammonia will begin to boil off at 192 deg. F.

## TABLE 16

### Table of Capacities and Surfaces in Tons of Refrigeration

Square feet of surface in Generator, Absorber, and Cooling Water per ton of Refrigeration under varying temperatures and pressures. Also Standard and Minimum steam pressure on Generator and factors for determining required ice making surfaces of Con., W. L. Coolers, Exch. and Rect. from given refrigerating capacity.

| Brine Temperature | | Cooling Water 55 Deg. F. | | | | | | | | | Cooling Water 60 Deg. F. | | | | | | | | | Tons R / Factor = tons of ice | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Generator | | | Absorber, square feet per ton | | | | | | Generator | | | Absorber, square feet per ton | | | | | | | | |
| | | Square feet per ton | Steam pressure | Pounds of steam per hour per ton | Horizontal tubular | Multipipe | Atmospheric | Dry only | D. P. | Gallons of water per minute per ton | Square feet per ton | Steam pressure | Pounds of steam per hour per ton | Horizontal tubular | Multipipe | Atmospheric | Dry only | D. P. | Gallons of water per minute per ton | Cond. and Ammonia pump | W. L. Cooler | Exch. and Rect. |
| 15 | Std. | 14.5 | 3.0 | 29 | 22 | 14 | 18 | 16 | 6.0 | 2.00 | 14.0 | 5.0 | 30 | 23 | 15 | 19 | 17 | 7.0 | 2.25 | | | |
| 0 | Min. | 15.5 | 1.0 | 30 | 23 | 15 | 19 | 17 | 6.5 | 2.25 | 15.0 | 3.0 | 31 | 24 | 16 | 20 | 18 | 7.5 | 2.50 | | | |
| 10 | Std. | 14.5 | 3.0 | 29 | 22 | 14 | 18 | 16 | 6.0 | 2.00 | 14.0 | 5.0 | 30 | 23 | 15 | 19 | 17 | 7.0 | 2.25 | | | |
| 0 | Min. | 15.5 | 1.5 | 30 | 23 | 15 | 19 | 17 | 6.5 | 2.25 | 15.0 | 3.0 | 31 | 24 | 16 | 20 | 18 | 7.5 | 2.50 | | | |
| 5 | Std. | 14.5 | 3.0 | 30 | 22 | 14 | 18 | 16 | 6.0 | 2.00 | 14.0 | 5.0 | 31 | 23 | 15 | 19 | 17 | 7.0 | 2.25 | 1.7 | 1.75 | 1.75 |
| 0 | Min. | 15.5 | 2.0 | 31 | 23 | 15 | 19 | 17 | 6.5 | 2.25 | 15.0 | 3.0 | 32 | 24 | 16 | 20 | 18 | 7.5 | 2.50 | | | |
| 0 | Std. | 14.5 | 3.5 | 30 | 23 | 15 | 19 | 17 | 6.5 | 2.25 | 14.0 | 5.0 | 31 | 24 | 16 | 20 | 18 | 7.5 | 2.25 | | | |
| | Min. | 15.5 | 2.5 | 31 | 24 | 16 | 20 | 18 | 7.0 | 2.50 | 15.0 | 3.5 | 32 | 25 | 17 | 21 | 19 | 8.0 | 2.50 | | | |
| 0 | Std. | 15.0 | 4.5 | 31 | 23 | 15 | 19 | 17 | 6.5 | 2.25 | 14.5 | 6.0 | 32 | 24 | 16 | 20 | 18 | 7.5 | 2.75 | | | |
| 5 | Min. | 16.0 | 3.0 | 32 | 24 | 16 | 20 | 18 | 7.0 | 2.50 | 15.5 | 4.5 | 33 | 25 | 17 | 21 | 19 | 8.0 | 2.50 | | | |
| 0 | Std. | 15.0 | 5.5 | 31 | 23 | 15 | 19 | 17 | 6.5 | 2.25 | 14.5 | 7.0 | 32 | 24 | 16 | 20 | 18 | 7.5 | 2.75 | | | |
| 10 | Min | 16.0 | 3.5 | 32 | 24 | 16 | 20 | 18 | 7.0 | 2.50 | 15.5 | 5.0 | 33 | 25 | 17 | 21 | 19 | 8.0 | 2.75 | | | |
| 0 | Std. | 15.0 | 6.0 | 32 | 24 | 16 | 20 | 18 | 7.0 | 2.50 | 14.5 | 8.0 | 33 | 25 | 17 | 21 | 19 | 8.0 | 3.00 | | | |
| 5 | Min. | 16.0 | 4.0 | 33 | 25 | 17 | 21 | 19 | 7.5 | 2.75 | 15.5 | 6.0 | 34 | 26 | 18 | 22 | 20 | 8.5 | 2.75 | | | |
| 0 | Std. | 15.0 | 7.5 | 33 | 6* | .. | .. | 19 | .. | 2.50 | 14.5 | 10.0 | 34 | 6* | .. | .. | 20 | .. | 3.00 | | | |
| 20 | Min. | 16.0 | 5.0 | 34 | 7* | .. | .. | 20 | .. | 2.75 | 15.5 | 7.5 | 35 | 7* | .. | .. | 21 | | | | | |

Water run in series Fig. Abs. 10 deg. higher. Water run in series Fig. 10 deg. higher on abs.

| Brine Temperature | | Cooling Water 65 Deg. F. | | | | | | | | | Cooling Water 70 Deg. F. | | | | | | | | | Tons R / Factor = tons of ice | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Generator | | | Absorber, square feet per ton | | | | | | Generator | | | Absorber, square feet per ton | | | | | | | | |
| | | Square feet per ton | Steam pressure | Pounds of steam per hour per ton | Horizontal tubular | Multipipe | Atmospheric | Dry only | D. P. | Gallons of water per minute per ton | Square feet per ton | Steam pressure | Pounds of steam per hour per ton | Horizontal tubular | Multipipe | Atmospheric | Dry only | D. P. | Gallons of water per minute per ton | Cond. and Ammonia pump | W. L. Cooler | Exch. and Rect. |
| 15 | Std. | 13.5 | 7 | 30 | 24 | 16½ | 20 | 18.5 | 8.0 | 2.75 | 13.0 | 9 | 31 | 26 | 18 | 21 | 20.5 | 9.5 | 3.25 | | | |
| 0 | Min. | 14.5 | 4 | 31 | 25 | 17½ | 21 | 19.5 | 8.5 | 3.00 | 14.0 | 6 | 32 | 27 | 19 | 22 | 21.5 | 10.0 | 3.50 | | | |
| 10 | Std. | 13.5 | 7 | 30 | 24 | 16½ | 20 | 18.5 | 8.0 | 2.75 | 13.0 | 9 | 31 | 26 | 18 | 21 | 20.5 | 9.5 | 3.25 | | | |
| 0 | Min. | 14.5 | 4 | 31 | 25 | 17½ | 21 | 19.5 | 8.5 | 3.00 | 14.0 | 6 | 32 | 27 | 19 | 22 | 21.5 | 10.0 | 3.50 | | | |
| 5 | Std. | 13.5 | 8 | 30 | 24 | 16½ | 20 | 18.5 | 8.0 | 2.75 | 13.0 | 10 | 31 | 26 | 18 | 22 | 20.5 | 9.5 | 3.25 | | | |
| 0 | Min. | 14.5 | 5 | 31 | 25 | 17½ | 21 | 19.5 | 8.5 | 3.00 | 14.0 | 7 | 32 | 27 | 19 | 23 | 21.5 | 10.0 | 3.50 | 1.75 | 1.8 | 1.8 |
| 0 | Std. | 13.5 | 8 | 31 | 25 | 17 | 21 | 19.0 | 8.5 | 3.00 | 13.0 | 10 | 32 | 27 | 19 | 22 | 21.0 | 10.0 | 3.50 | | | |
| | Min. | 14.5 | 5 | 32 | 26 | 18 | 22 | 20.0 | 9.0 | 3.25 | 14.0 | 7 | 33 | 28 | 20 | 23 | 22.0 | 10.5 | 3.75 | | | |
| 0 | Std. | 14.0 | 9 | 31 | 25 | 17 | 21 | 19.0 | 8.5 | 3.00 | 13.0 | 12 | 32 | 27 | 19 | 22 | 21.0 | 10.0 | 3.50 | | | |
| 5 | Min. | 15.0 | 6 | 32 | 26 | 18 | 22 | 20.0 | 9.0 | 3.25 | 14.5 | 9 | 33 | 28 | 20 | 23 | 22.0 | 10.5 | 3.75 | | | |
| 0 | Std. | 14.0 | 10 | 32 | 25 | 18 | 22 | 19.0 | 9.0 | 3.25 | 13.5 | 13 | 33 | 27 | 20 | 23 | 22.0 | 10.5 | 3.50 | | | |
| 10 | Min. | 15.0 | 7 | 33 | 26 | 19 | 23 | 20.0 | 9.5 | 3.50 | 14.5 | 10 | 34 | 28 | 21 | 24 | 23.0 | 11.0 | 3.75 | | | |
| 0 | Std. | 14.0 | 12 | 32 | 26 | 18 | 22 | 20.0 | 9.0 | 3.25 | 13.5 | 14 | 33 | 6* | 20 | 23 | 22.0 | 11.0 | 3.75 | | | |
| 15 | Min. | 15.0 | 8 | 33 | 27 | 19 | 23 | 21.0 | 9.5 | 3.50 | 14.5 | 11 | 34 | 7* | 21 | 24 | 23.0 | 11.5 | 4.00 | | | |
| 0 | Std. | 14.0 | 14 | 33 | 6* | .. | .. | .. | .. | 3.50 | 13.5 | 15 | 34 | 6* | .. | .. | .. | .. | 3.75 | | | |
| 20 | Min. | 15.0 | 10 | 34 | 7* | .. | .. | .. | .. | 3.75 | 14.5 | 12 | 35 | 7* | .. | .. | .. | .. | 4.00 | | | |

* Use Dry Abs. of surf. indicated in connection with Hor. Tub. Surf. for −15 deg. brine.

TABLE 16—*Continued*

Always to be used for standard ice making plants.

| Brine Temperature | | Cooling Water 75 Deg. F. | | | | | | | | | Cooling Water 80 Deg. F. | | | | | | | | | Tons R / Factor = tons of ice | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Generator | | | Absorber, square feet per ton | | | | | Gallons of water per minute per ton | Generator | | | Absorber, square feet per ton | | | | | Gallons of water per minute per ton | | | |
| | | Square feet per ton | Steam pressure | Pounds of steam per hour per ton | Horizontal tubular | Multipipe | Atmospheric | Dry only | D. P. | | Square feet per ton | Steam pressure | Pounds of steam per hour per ton | Horizontal tubular | Multipipe | Atmospheric | Dry only | D. P. | | Cond. and Ammonia pump | W. L. Cooler | Exch. and Rect. |
| 15 | Std. | 12.5 | 10 | 31 | 28 | 20 | 22 | 23 | 11.0 | 3.75 | 13.0 | 15 | 32 | 31 | 22 | 24 | 26 | 13.0 | 4.75 | | | |
| 0 | Min. | 13.5 | 8 | 32 | 29 | 21 | 23 | 24 | 11.5 | 4.00 | 14.0 | 13 | 33 | 32 | 23 | 25 | 27 | 13.5 | 5.00 | | | |
| 10 | Std. | 12.5 | 10 | 31 | 28 | 20 | 22 | 23 | 11.0 | 3.75 | 13.0 | 15 | 32 | 31 | 22 | 24 | 26 | 13.0 | 4.75 | | | |
| 0 | Min. | 13.5 | 8 | 32 | 29 | 21 | 23 | 24 | 11.5 | 4.00 | 14.0 | 13 | 33 | 32 | 23 | 25 | 27 | 13.5 | 5.00 | | | |
| 5 | Std. | 12.5 | 10 | 32 | 28 | 20 | 23 | 23 | 11.0 | 4.00 | 13.0 | 15 | 33 | 32 | 22 | 24 | 26 | 13.5 | 5.00 | 1.82 | 1.85 | 1.85 |
| 0 | Min. | 13.5 | 8 | 33 | 29 | 21 | 24 | 24 | 11.5 | 4.25 | 14.0 | 13 | 34 | 33 | 23 | 25 | 27 | 14.0 | 5.25 | | | |
| 0 | Std. | 13.0 | 12 | 32 | 29 | 21 | 23 | 24 | 11.5 | 4.00 | 13.0 | 18 | 33 | 32 | 23 | 25 | 27 | 13.5 | 5.00 | | | |
| | Min. | 14.0 | 9 | 33 | 30 | 22 | 24 | 25 | 12.0 | 4.25 | 14.0 | 15 | 34 | 33 | 24 | 26 | 28 | 14.0 | 5.25 | | | |
| 0 | Std. | 13.0 | 12 | 32 | 29 | 21 | 24 | 24 | 11.5 | 4.25 | 13.5 | 20 | 33 | 33 | 23 | 25 | 27 | 14.0 | 5.25 | | | |
| 5 | Min. | 14.0 | 10 | 33 | 30 | 22 | 25 | 25 | 12.0 | 4.50 | 14.5 | 16 | 34 | 34 | 24 | 26 | 28 | 14.5 | 5.50 | | | |
| 0 | Std. | 13.0 | 14 | 33 | 30 | 22 | 24 | 25 | 12.0 | 4.25 | 13.5 | 21 | 34 | 33 | 24 | 26 | 28 | 14.5 | 5.50 | | | |
| 10 | Min. | 14.0 | 12 | 34 | 31 | 23 | 25 | 26 | 12.5 | 4.50 | 14.5 | 17 | 35 | 34 | 25 | 27 | 29 | 15.0 | 5.75 | | | |
| 0 | Std. | 13.5 | 15 | 34 | 6* | 23 | 25 | 25 | 12.5 | 4.50 | 14.0 | 22 | 35 | 6* | .. | .. | .. | .. | 5.75 | | | |
| 15 | Min. | 14.5 | 13 | 35 | 7* | 24 | 26 | 26 | 13 | 04.75 | 15.0 | 18 | 36 | 7* | .. | .. | .. | .. | 6.00 | | | |
| 0 | Std. | 13.5 | 17 | 35 | 6* | .. | .. | .. | .. | 4.75 | 14.0 | 24 | 36 | 6* | .. | .. | .. | .. | 6.00 | | | |
| 20 | Min. | 14.5 | 15 | 36 | 7* | .. | .. | .. | .. | 5.00 | 15.0 | 20 | 37 | 7* | .. | .. | .. | .. | 6.25 | | | |

Water run in series Fig. Abs. 10 deg. higher. Water 80 deg. to Absorber.

| Brine Temperature | | Cooling Water 85 Deg. F. | | | | | | | | | Cooling Water 90 Deg. F. | | | | | | | | | Tons R / Factor = tons of ice | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Generator | | | Absorber, square feet per ton | | | | | Gallons of water per minute per ton | Generator | | | Absorber, square feet per ton | | | | | Gallons of water per minute per ton | | | |
| | | Square feet per ton | Steam pressure | Pounds of steam per hour per ton | Horizontal tubular | Multipipe | Atmospheric | Dry only | D. P. | | Square feet per ton | Steam pressure | Pounds of steam per hour per ton | Horizontal tubular | Multipipe | Atmospheric | Dry only | D. P. | | Cond. and Ammonia pump | W. L. Cooler | Exch. and Rect. |
| 15 | Std. | 13.0 | 20 | 33 | 34 | 25 | 26 | 30 | 15 | 6.0 | 13.5 | 28 | 34 | 39 | 29 | 30 | 35 | 18 | 7.5 | | | |
| 0 | Min. | 14.0 | 18 | 34 | 35 | 26 | 27 | 31 | 16 | 6.5 | 14.5 | 25 | 35 | 40 | 40 | 31 | 36 | 19 | 8.0 | | | |
| 10 | Std. | 13.0 | 20 | 33 | 34 | 25 | 26 | 30 | 15 | 6.0 | 13.5 | 28 | 34 | 39 | 29 | 30 | 35 | 18 | 7.5 | | | |
| 0 | Min. | 14.0 | 18 | 34 | 35 | 26 | 27 | 31 | 16 | 6.5 | 14.5 | 24 | 35 | 40 | 30 | 31 | 36 | 19 | 8.0 | | | |
| 5 | Std. | 13.5 | 21 | 34 | 34 | 26 | 26 | 31 | 15 | 6.5 | 13.5 | 30 | 35 | 39 | 30 | 31 | 36 | 18 | 8.0 | | | |
| 0 | Min. | 14.5 | 18 | 35 | 35 | 27 | 27 | 32 | 16 | 7.0 | 14.5 | 26 | 36 | 40 | 31 | 32 | 37 | 19 | 8.5 | | | |
| 0 | Std. | 13.5 | 23 | 35 | 35 | 26 | 27 | 31 | 16 | 6.5 | 14.0 | 32 | 36 | 40 | 30 | 31 | 36 | 19 | 8.0 | | | |
| | Min. | 14.5 | 20 | 36 | 36 | 27 | 28 | 32 | 17 | 7.0 | 15.0 | 28 | 37 | 41 | 31 | 32 | 37 | 20 | 8.5 | 1.9 | 1.9 | 1.9 |
| 0 | Std. | 14.0 | 25 | 36 | 35 | 27 | 27 | 32 | 16 | 7.0 | 14.0 | 34 | 37 | 40 | 31 | 32 | 37 | 19 | 8.5 | | | |
| 5 | Min. | 15.0 | 22 | 37 | 36 | 28 | 28 | 33 | 17 | 7.5 | 15.0 | 30 | 38 | 41 | 32 | 33 | 38 | 20 | 9.0 | | | |
| 0 | Std. | 14.0 | 30 | 37 | 36 | 27 | 28 | 32 | 17 | 7.0 | 14.5 | 40 | 38 | 41 | 33 | 33 | 37 | 20 | 8.5 | | | |
| 10 | Min. | 15.0 | 26 | 38 | 37 | 28 | 29 | 33 | 18 | 7.5 | 15.5 | 36 | 39 | 42 | 34 | 34 | 38 | 21 | 9.0 | | | |

Water 85 deg. to Absorber. Water 90 deg. to Absorber.

* Use Dry Abs. of surf. indicated in connection with Hor. Tub. surf. for − 10 deg. brine.

TABLE 16—*Continued*

| Temperature of Water, Degrees.. | | 55 | 60 | 65 | 70 | 75 | 80 | 85 | 90 |
|---|---|---|---|---|---|---|---|---|---|
| Type | | Surface, Square Feet per Ton | | | | | | | |
| Condensers: | | | | | | | | | |
| Good water ① | | | | | | | | | |
| Double pipe..... | Ice | 15.0 | 16.5 | 18.0 | 19.5 | 21.0 | 23.0 | 25.0 | 28.0 |
| | Ref. | 8.8 | 9.7 | 10.3 | 11.3 | 11.5 | 12.6 | 13.2 | 14.7 |
| Bad water ① | | | | | | | | | |
| Atmospheric..... | Ice | ..... | 44.0 | 47.0 | 50.0 | 53.0 | 57.0 | 61.0 | 65.0 |
| | Ref. | ..... | 26.0 | 27.0 | 28.2 | 29.6 | 31.2 | 33.0 | 35.0 |
| Good water ② | | | | | | | | | |
| Horizontal tabular..... | Ice | 30.0 | 34.0 | 38.0 | 42.0 | 46.0 | 50.0 | 54.0 | 60.0 |
| Where have less room | Ref. | 17.6 | 20.0 | 21.7 | 24.0 | 25.2 | 27.5 | 29.4 | 31.6 |
| Good water ③ | | | | | | | | | |
| Multitube..... | Ice | 17.0 | 18.5 | 20.0 | 21.5 | 23.0 | 25.0 | 27.0 | 30.0 |
| | Ref. | 10.0 | 10.9 | 11.4 | 12.3 | 12.6 | 13.8 | 14.2 | 15.8 |
| Weak liquor cooler: | | | | | | | | | |
| Good water ① | | | | | | | | | |
| Double pipe..... | Ice | 5.0 | 5.5 | 6.0 | 6.5 | 7.0 | 7.5 | 8.3 | 9.3 |
| | Ref. | 2.9 | 3.3 | 3.5 | 3.8 | 4.1 | 4.4 | 4.9 | |
| Bad water ① | | | | | | | | | |
| Atmospheric..... | Ice | ..... | 11.0 | 11.75 | 12.50 | 13.25 | 14.25 | 15.25 | 16.25 |
| | Ref. | ..... | 6.5 | 6.9 | 7.35 | 7.8 | 8.4 | 9.0 | 9.6 |
| Good water ② | | | | | | | | | |
| Multitube..... | Ice | 5.66 | 6.16 | 6.66 | 7.16 | 7.66 | 8.33 | 9.0 | 10.0 |
| Where less space | Ref. | 3.33 | 3.62 | 3.80 | 4.1 | 4.2 | 4.58 | 4.74 | 5.26 |
| Distilled water cooler: | | | | | | | | | |
| Double pipe..... | Ice | 5.5 | 6.0 | 6.4 | 6.8 | 7.2 | 7.6 | 8.0 | 8.5 |

| Horizontal multipass brine cooler... | Brine Temperature, Degrees F. Range = Approximately 10 Deg. | | | | |
|---|---|---|---|---|---|
| | +10 | 0 | −5 | −10 | −15 |
| | Surface per Ton Refrigeration | | | | |
| | 11 | 12 | 14 | 15 | 16 |

NOTE

For single pass brine coolers for ice making tanks, use 35 sq. ft. per ton ice.

| | | |
|---|---|---|
| Exchanger..... | Double pipe = 9.2 sq. ft. per ton ice.<br>Double pipe = 5.3 sq. ft. per ton ref. | Takes up more room.<br>Use on units of 25 tons ice or less. |
| | Multitube = 10.56 sq. ft. per ton ice.<br>Multitube = 5.86 sq. ft. per ton ref. | Takes up less room.<br>Use on units over 25 tons ice. |
| Rectifier..... | Double pipe = 4.86 sq. ft. per ton ice.<br>Double pipe = 2.85 sq. ft. per ton ref. | Same as for Exchanger. |
| | Multitube = 6.6 sq. ft. per ton ice.<br>Multitube = 3.9 sq. ft. per ton ref. | Same as for Exchanger. |

TABLE 16—*Continued*

AMMONIA PUMPS

| Capacity, Tons | | Size | Capacity in Pounds at 28 Deg. B. Aqua Ammonia per Minute, 36 r.p.m. | Floor Space, Inches |
|---|---|---|---|---|
| Ice | Ref. | St. Cyl. $NH_3$ Cyl. Stroke | | |
| 7 | 13 | 6 × $2\frac{3}{4}$ × 8 | 26.2 | 12×56 |
| 10 | 18 | 8 × $3\frac{1}{4}$ × 8 | 63.4 | 12×56 |
| 12 | 21 | 8 × $3\frac{1}{2}$ × 8 | 75.1 | 12×56 |
| 15 | 26 | 8 × $3\frac{1}{4}$ × 12 | 92.2 | 19×80 |
| 18 | 31 | 8 × $3\frac{1}{2}$ × 12 | 108.7 | 19×80 |
| 25 | 46 | 10 × $4\frac{1}{4}$ × 12 | 163.4 | 19×81 |
| 30 | 52 | 10 × $4\frac{1}{2}$ × 12 | 184.9 | 19×81 |
| 40 | 73 | 12 × $5\frac{1}{4}$ × 12 | 256.6 | 20×81 |
| 45 | 80 | 12 × $5\frac{1}{2}$ × 12 | 283.0 | 20×81 |
| 50 | 96 | 14 × 6 × 12 | 339.5 | 21×83 |
| 60 | 114 | 14 × $6\frac{1}{2}$ × 12 | 400.8 | 21×83 |
| 100 | 200 | 16 × $7\frac{1}{2}$ × 16 | 705.2 | 27×97 |
| 130 | 228 | 16 × 8 × 16 | 806.4 | 27×97 |
| 130 | 228 | 18 × 8 × 16 | 806.4 | 29×97 |

The total heat absorbed, per 1.0 lb. of ammonia boiled off, = 1547.2 B.t.u. In the rectifier the water content will be reduced to 0.2 per cent and the difference, 7.2 per cent, will be condensed and will form a strong aqua of 68 per cent concentration. There will be, then, a return to the generator of 15.3 per cent of ammonia, and the value 1547.2 [neglecting the action of the strong aqua] must be increased to $\frac{1547.2}{0.847}$ = 1827 B.t.u. Under standard conditions of operation (5 deg. F. and 145 lb. abs. in this case) 0.412 lb. of ammonia per minute are required or 0.412 × 1827 = 753 B.t.u. If each pound of steam supplies 964 B.t.u. then $\frac{753 \times 60}{964}$ = 46.9 lb. of steam at 6.0 lb. gage pressure would be required per ton of refrigeration per hour, an amount somewhat greater than that given in the tables of capacities and surfaces and used in the specification of absorption machines. It will be noted that the value 46.9 can be reduced by using a greater difference in the concentrations either by means of a higher steam pressure or a colder cooling water in the absorber. The fact that water boils as well as ammonia causes some operating difficulties and necessitates the use of a rectifier which functions as a precooler but is carefully operated so as to remove part and not all of the superheat, thereby reducing the temperature of the gas to a point where the greater part of the water vapor will condense out. This action as well as the action of the entire cycle is well described in Bulletin No. 146 of the Engineering Experiment Station, University of Illinois, a part of which is abstracted in the following.

## APPLICATION TO AMMONIA ABSORPTION PROCESS

[From Bulletin No. 146, University of Illinois Experiment Station]

**Problem.**—The use of the tables and diagrams can be brought out best by a study of the operation of an ammonia absorption refrigerating machine. In order to avoid indefinite generalities a particular case will be taken.

In this problem the ammonia solution enters the generator with a concentration of 35 per cent, and leaves after the boiling off of the ammonia with a reduced concentration of 25 per cent. The condenser pressure is 150 lb. per square inch absolute, which will be taken as the generator pressure likewise. The average concentration in the generator may be considered as 30 per cent. The given generator pressure will be reached when the solution has been heated to 217.4 deg. F. (Fig. 60) by the steam admitted to the heating coils. Under these conditions the vapors leaving the generator will show a water content of 7.4 per cent, corresponding to a partial pressure of 11.10 lb. for the water vapor and 138.9 lb. for the ammonia.

On passing through the rectifier the vapors undergo partial condensation by cooling. The usual practice is to cool the vapors to a temperature approximately 20 degrees above the condensation point of anhydrous ammonia at the pressure in use. In the present case the vapors will be cooled to a temperature of 98.81 degrees as will be seen by reference to the tables of the Bureau of Standards.[8] This corresponds to a concentration of ammonia in the solution phase of 71.3 per cent, and (for a temperature of 98.8 deg. F. and a concentration of ammonia of 71.3 per cent) a concentration of water in the vapor phase of 0.160 per cent. In other words, in their passage through the rectifier the vapors have been robbed of part of their contained water, the partial pressure of which has been reduced to 0.25 lb., while that of the ammonia has been increased to 149.75 lb.

To do this some of the ammonia vapor evolved in the generator has been sacrificed. It is impossible to return any of the original water in the vapor to the generator without returning such an amount of ammonia that the concentration of the solution formed will be in equilibrium with the vapors remaining after the condensation. Likewise it is impossible to cause this condensation without a cooling of the solution, unless the vapor pressure be materially increased.

**Solution of Problem.**—Consideration of the case of 2000 pounds of vapor mixture in the rectifier after leaving the generator will show the extent of this process:

| | | |
|---|---|---|
| Water content of vapor from generator | 7.4 | per cent |
| Weight of water | 148.0 | lb. |
| Weight of ammonia | 1852.0 | lb. |
| Ammonia content of solution returned to generator from rectifier | 71.3 | per cent |
| Weight of water returned | 144.8 | lb. |
| Weight of ammonia | 359.7 | lb. |
| Total | 504.5 | lb. |
| Water content of gases sent on to receivers. | 0.160 | per cent |
| Weight of water passed | 2.393 | lb. |
| Weight of ammonia | 1493.10 | lb. |
| Total | 1495.50 | lb. |
| Partial pressure of water vapor | 0.25 | lb. per square inch |
| Partial pressure of ammonia | 149.75 | lb. per square inch |

[8] B. S. Circular No. 142, 1923.

On flowing through the pressure reducing, or so-called expansion, valve the ammonia solution experiences a release in pressure to a value determined by the concentration of the aqua ammonia in the absorber and by the temperature of the latter. Since the pressure will be greater the higher the concentration, and since no concentration changes occur during the pumping of the solution from the absorber to the generator, it follows that the back pressure in the cooling coils will be determined in the present instance by the vapor pressure of a 35 per cent ammonia solution at whatever temperature the absorber may be kept during the operation of the machine.

It is desired to maintain a temperature of 6 deg. F. in the cooling coils. Although further experimental work concerning these lower temperature vapor pressures must be done before the tables can be extended into this region, a fairly satisfactory extrapolation can be performed by the use of the aqueous vapor content table. There it will be noticed that as a general rule an increase in temperature of approximately 41.0 deg. F. will cause an increase of about 5 per cent in the composition of the solution with which a given composition of the vapor phase is in equilibrium. This rule may be applied in order to calculate from the data furnished in the table for a temperature of 32 deg. F. the solution with which a given vapor phase will be in equilibrium at 6 deg. F. The quantity calculated in this manner as the amount to be either subtracted from or added to the values there tabulated may be taken as 3.17 per cent.

A vapor possessing a water content of 0.160 per cent is in equilibrium with a 54.1 per cent ammonia solution at 32 deg. F. Applying the preceding, this same vapor will be in equilibrium with a 50.9 per cent solution at 6 deg. F. The value of the quotient corresponding to a solution of this concentration is 0.903. Knowing the temperature of the solution in the coils and the quotient, the value of $\Theta$ may be calculated, and is found to be 421.0 degrees or a Fahrenheit temperature of $-$ 39.0. The tables of the Bureau of Standards show a pressure of 10.72 lb. of ammonia at this temperature, which will be reached by a 35 per cent ammonia solution at a temperature of 51.3 degrees. In other words, since at this temperature the composition of the vapor required in the coils is identical with the composition of the solution furnished at the expansion valve, there should be no accumulation of a liquid phase in the freezing coils; and, of course, the same condition is true for any absorber temperature below this, provided only that the process occurs at constant pressure.

On the other hand the tables of the Bureau of Standards give the value of 35.09 lb. for anhydrous ammonia at 6 deg. F. This is the pressure attained by a 35 per cent ammonia solution at 105.9 deg. F. and at this pressure the solution phase is identical in composition with that furnished by the expansion valve. No evaporation of water will therefore occur in the freezing coils. In order that the machine may work properly the average temperature of the absorber must be kept below the value of 105.9 deg. F.

We may assume that the absorber temperature is that of the solution leaving the absorber, namely, 80 deg. F. The pressure is 20.5 lb. and the quotient is now 0.954, corresponding to a concentration of solution of 66.1 per cent at 6 deg. F. The corresponding concentration at 32 deg. F. is 69.3 per cent, according to the suggested method, in equilibrium with a vapor of 0.065 per cent water content. The amount of water appearing as an accumulating liquid phase is evidently represented by the difference between the water content of the liquid phase as supplied by the valve and that of the vapor, or 0.095 per cent. Of the 2.39 lb. of water passing

the valve 1.42 lb. will accumulate in the coils to form 4.19 lb. of 66.1 per cent solution, and the rest will evaporate off into the vapor phase together with the residual ammonia.

**Performance.**—On the basis of 60,700 lb. of ammonia representing the quantity which must be evaporated to supply a refrigerating effect of 100 tons per day, Table 17*b*, the table of preformance, was constructed.

TABLE 17*a*

TABLE OF PERFORMANCE FOR ABSORPTION REFRIGERATING MACHINE

| | Pounds per 2000 Lb. of Vapor | Pounds per 100 Tons of Refrigeration per 24 Hours |
|---|---|---|
| Weight of original vapor | 2000.00 | 81,300.0 |
| Containing water | 148.00 | 6,020.0 |
| And ammonia | 1852.00 | 75,280.0 |
| Returned to generator, water | 144.8 | 5,890.0 |
| Returned to generator, ammonia | 359.7 | 14,630.0 |
| Sent to receiver, water | 2.39 | 97.2 |
| Sent to receiver, ammonia | 1493.1 | 60,700.0 |
| Accumulated in coils, water | 1.42 | 57.7 |
| Accumulated in coils, ammonia | 2.77 | 112.6 |
| Condensed by absorber, water | 0.97 | 39.5 |
| Condensed by absorber, ammonia | 1490.3 | 60 582.0 |

**Effect of Water Content.**—At this point it may be well to clear up an apparent misconception with regard to the nature of the refrigerating medium in the absorption process. In the compression process anhydrous ammonia itself is the substance affecting the essential heat transfers. In the absorption process this is not the case, but a concentrated solution of ammonia, or better still, a solution of water in ammonia is used as the active agent. This itself in no way affects the value of the machine for producing cold; in fact, as far as can be predicted with our present knowledge of the ammonia solution, this should increase its efficiency since the latent heat of water boiling under any pressure is much higher than that of ammonia at the same pressure. The specific heats of both liquid and vapor of steam and of ammonia are nearly identical. On the other hand, the use of a solution in the coils instead of a pure substance introduces the undesirable effect of selective evaporation which under ordinary working conditions causes an accumulation of a certain amount of liquid phase in the freezing pipes. Probably most of this is eventually carried along to the absorber by the impetus of the vapor current, thus preventing clogging. It must be

remembered that the calculation here appended concerns only constant pressure processes. A pressure drop through the freezing coils will affect these calculations unfavorably by further increasing the amount which will form of this liquid phase.

Furthermore, it is evident now that this condensation does not form water but a concentrated ammonia solution in the coils. Herein lies the explanation of the fact that these machines can have condensation occurring in the coils and yet not suffer plugging. Although no work has been done to determine the freezing points of ammonia solutions, the law regarding the lowering of the freezing point is universally true; and under the conditions of concentration existing in the cooling coil it would be impossible to plug the line with frozen solution unless the back pressure from the absorber were materially reduced.

**Operating Conditions.**—In the operating conditions which we have just been considering there is another interesting possibility to be investigated. In the analysis it was stated that the vapors passing through the rectifier were cooled to a temperature of 98.81 degrees, or 20 degrees above the condensation point of anhydrous ammonia. It is at once evident that the amount of accumulated liquid in the brine coils will be materially decreased if this rectifying is carried still further; and it may not be out of place to study the operation of a machine in which the vapors leave the rectifier at a temperature of 88.81 degrees or 10 degrees below the first rectifying temperature.

To save space the author has appended Table 17*b* as a means of contrasting the results due to the two operating conditions. The table is self-explanatory. It is evident that in order to secure identical refrigerating effects, the quantity of vapor furnished by the generator must be much greater in the second case than in the first, and a much larger amount of ammonia must be returned to the generator from the rectifier. On the other hand, it is possible to raise the absorber to a higher temperture before accumulation begins in the freezing coils (optimum temperature), and as would be expected the amount of this accumulation is not so great for any given temperature, thus increasing the refrigerating efficiency of the coils. Yet the maximum temperature to which the absorber can be raised without prohibiting the proper functioning of the machine is not altered.

Here again the engineer is confronted by conflicting conditions between which it is better to effect a compromise, but the nature of this compromise must be determined by the results to be obtained. A concentrated ammonia solution boils at a lower temperature than one more dilute; hence it follows that in plants desiring the attainment of a cold temperature fairly close to the bottom of the range capable of being reached with the absorption machine it would be better to keep the rectifier at a lower temperature, and thus furnish the freezing coils with the more concentrated ammonia. In those localities in which the climate is such that the cooling water around the absorber does not keep its temperature very low, it would greatly increase the working efficiency of the machine to supply only the more concentrated ammonia to the coils; but otherwise there seems to be no practical advantage in keeping the rectifier at this lower temperature, in view of the heavier load which is thus transferred to the generator.

The chart, Fig. 60, will be found very useful in the solution of engineering problems. From it values required for computations can be found directly, accurately enough for most work. In addition a clearer idea of conditions can generally be obtained from a chart than by the use of tables, and this is particularly true in the case of the absorption machine cycle.

TABLE 17*b*

Comparative Performance for Different Conditions of Operation

| Items | Per Ton of Vapor | Per 100 Tons of Refrigeration per 24 Hours | Per Ton of Vapor | Per 100 Tons of Refrigeration per 24 Hours |
|---|---|---|---|---|
| Weight of original vapor (pounds) | 2000.0 | 81,300 | 2000.0 | 115,200 |
| Weight of contained water | 148.0 | 6,020 | 148.0 | 8,500 |
| Weight of contained ammonia | 1852.0 | 75,280 | 1852.0 | 106,700 |
| Temperature of vapor leaving rectifier (degrees F.) | 98.81 | | 88.81 | |
| Concentration of solution returned to generator (per cent) | 71.30 | | 84.50 | |
| Weight returned (pounds) | 504.5 | 20,530 | 946.0 | 54,490 |
| Weight of water | 144.8 | 5,890 | 146.6 | 8,440 |
| Weight of ammonia | 359.7 | 14,630 | 799.4 | 46,050 |
| Water content of vapors from receivers (per cent) | 0.160 | | 0.065 | |
| Weight of this vapor (pounds) | 1495.5 | 60,800 | 1054.0 | 60,740 |
| Weight of water | 2.39 | 97.2 | 0.69 | 40 |
| Weight of ammonia | 1493.1 | 60,700 | 1053.3 | 60,700 |
| Maximum absorber temp. (degrees F.) | 105.60 | | 105.60 | |
| Maximum absorber pressure (pounds) | 35.09 | | 35.09 | |
| Optimum absorber temp. (degrees F.) | 51.30 | | 79.50 | |
| Optimum absorber pressure (pounds) | 10.72 | | 20.34 | |
| Pressure of absorber at 80 deg. F. (pounds per square inch) | 20.50 | | 20.50 | |
| Composition of equilibrium solution at 6 deg. F. (per cent) | 66.10 | | 66.10 | |
| Water content of vapor at 6 deg. F. (per cent) | 0.065 | | 0.065 | |
| Water content accumulating in coils per cent) | 0.095 | | 0.00 | |
| Weight of accumulating solution (pounds) | 4.19 | 170.3 | 0.0 | 0.0 |
| Weight of water | 1.42 | 57.7 | 0.0 | 0.0 |
| Weight of ammonia | 2.77 | 112.6 | 0.0 | 0.0 |
| Weight of vapor condensed in absorber (pounds) | 14.9130 | 60,621.3 | 1054.0 | 60,740.0 |
| Weight of water | 0.97 | 39.5 | 0.69 | 40.0 |
| Weight of ammonia | 1490.30 | 60,581.8 | 1053.3 | 60,700.0 |

TABLE 18

TOTAL VAPOR PRESSURES OF AQUA AMMONIA

Pressures are in Pounds per Square Inch Absolute

| Temp. Deg. F. | Molal Concentration of Ammonia in the Solutions in Percentages | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | 0 | 5 | 10 | 15 | 20 | 25 | 30 | 35 | 40 | 45 | 50 |
| 32 | 0.09 | 0.34 | 0.60 | 0.97 | 1.58 | 2.60 | 4.20 | 6.54 | 9.93 | 14.18 | 19.40 |
| 40 | 0.12 | 0.45 | 0.77 | 1.24 | 2.01 | 3.25 | 5.21 | 8.06 | 12.05 | 17.20 | 23.39 |
| 50 | 0.18 | 0.64 | 1.05 | 1.65 | 2.67 | 4.29 | 6.75 | 10.35 | 15.34 | 21.65 | 29.26 |
| 60 | 0.26 | 0.86 | 1.42 | 2.21 | 3.51 | 5.55 | 8.65 | 13.22 | 19.30 | 27.05 | 36.26 |
| 70 | 0.36 | 1.17 | 1.84 | 2.90 | 4.56 | 7.13 | 11.01 | 16.56 | 24.05 | 33.39 | 44.42 |
| 80 | 0.51 | 1.52 | 2.43 | 3.76 | 5.85 | 9.06 | 13.86 | 20.61 | 29.69 | 40.96 | 54.08 |
| 90 | 0.70 | 2.02 | 3.15 | 4.83 | 7.43 | 11.40 | 17.23 | 25.48 | 36.34 | 49.82 | 65.32 |
| 100 | 0.95 | 2.62 | 4.05 | 6.13 | 9.34 | 14.22 | 21.32 | 31.16 | 44.12 | 59.99 | 78.30 |
| 110 | 1.27 | 3.34 | 5.14 | 7.72 | 11.64 | 17.58 | 26.07 | 37.81 | 53.16 | 71.87 | 93.19 |
| 120 | 1.69 | 4.27 | 6.46 | 9.63 | 14.42 | 21.54 | 31.69 | 45.62 | 63.59 | 85.33 | 110.20 |
| 130 | 2.22 | 5.38 | 8.07 | 11.91 | 17.67 | 26.20 | 38.25 | 54.55 | 75.55 | 100.86 | 129.50 |
| 140 | 2.89 | 6.70 | 9.98 | 14.63 | 21.49 | 31.54 | 45.73 | 64.78 | 89.19 | 118.24 | 151.30 |
| 150 | 3.72 | 8.29 | 12.23 | 17.81 | 26.00 | 37.81 | 54.43 | 76.61 | 104.65 | 138.10 | 175.40 |
| 160 | 4.74 | 10.16 | 14.92 | 21.54 | 31.16 | 45.02 | 64.25 | 89.88 | 122.10 | 160.20 | 202.70 |
| 170 | 5.99 | 12.41 | 18.01 | 25.87 | 37.11 | 53.27 | 75.55 | 104.84 | 141.75 | 185.10 | 233.20 |
| 180 | 7 51 | 15.00 | 21.65 | 30.86 | 44.02 | 62.68 | 88.17 | 121.68 | 163.70 | 212.60 | 267.00 |
| 190 | 9.34 | 18.06 | 25.87 | 36.60 | 51.81 | 73.32 | 102.56 | 140.75 | 188.10 | 243.30 | 304.30 |
| 200 | 11.53 | 21.60 | 30.72 | 43.14 | 60.62 | 85.33 | 118.68 | 161.81 | 215.20 | 277.00 | 345.50 |
| 210 | 14.12 | 25.61 | 36.26 | 50.58 | 70.72 | 98.80 | 136.42 | 185.10 | 245.10 | 314.50 | 390.70 |
| 220 | 17.19 | 30.27 | 42.47 | 59.00 | 81.91 | 113.81 | 156.41 | 211.24 | 278.20 | 355.10 | 439.60 |
| 230 | 20.78 | 35.59 | 49.60 | 68.46 | 94.43 | 130.64 | 178.28 | 239.70 | 314.50 | 400.20 | 493.40 |
| 240 | 24.97 | 41.52 | 57.65 | 78.91 | 108.60 | 149.20 | 202.74 | 270.92 | 354.10 | 448.90 | 552.30 |
| 250 | 29.83 | 48.32 | 66.67 | 90.74 | 124.08 | 169.48 | 229.62 | 305.60 | 397.60 | 502.40 | |

TABLE 18—*Continued*

TOTAL VAPOR PRESSURES OF AQUA AMMONIA

Pressures are in Pounds per Square Inch Absolute

| Temp. Deg. F. | Molal Concentration of Ammonia in the Solutions in Percentages | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | 55 | 60 | 65 | 70 | 75 | 80 | 85 | 90 | 95 | 100 |
| 32 | 25.16 | 31.16 | 36.77 | 42.72 | 45.94 | 49.28 | 52.14 | 54.90 | 58.01 | 62.29 |
| 40 | 30.20 | 37.20 | 43.73 | 49.60 | 54.43 | 58.33 | 61.64 | 64.78 | 68.32 | 73.32 |
| 50 | 37.54 | 45.93 | 53.85 | 60.87 | 66.67 | 71.29 | 75.25 | 79.07 | 83.41 | 89.19 |
| 60 | 46.23 | 56.32 | 65.90 | 74.06 | 80.96 | 86.49 | 91.08 | 95.69 | 100.66 | 107.60 |
| 70 | 56.44 | 68.46 | 79.54 | 89.36 | 97.51 | 104.08 | 109.60 | 114.86 | 120.63 | 128.80 |
| 80 | 68.19 | 82.55 | 95.69 | 107.20 | 116.54 | 124.30 | 130.64 | 136.40 | 143.72 | 153.00 |
| 90 | 81.91 | 98.61 | 114.02 | 127.42 | 138.34 | 147.15 | 154.56 | 161.81 | 169.76 | 180.60 |
| 100 | 97.68 | 117.17 | 135.01 | 150.50 | 163.16 | 173.40 | 182.10 | 190.22 | 199.22 | 211.90 |
| 110 | 115.70 | 138.10 | 158.84 | 176.54 | 191.15 | 203.26 | 212.89 | 222.34 | 232.85 | 247.00 |
| 120 | 136.20 | 162.08 | 185.70 | 206.29 | 222.68 | 236.37 | 247.38 | 258.40 | 270.10 | 286.40 |
| 130 | 159.40 | 189.00 | 215.88 | 239.33 | 258.40 | 273.30 | 286.40 | 298.67 | 311.90 | 330.30 |
| 140 | 185.40 | 219.28 | 249.66 | 276.15 | 297.81 | 315.00 | 329.40 | 343.20 | 358.60 | 379.10 |
| 150 | 214.50 | 252.65 | 287.24 | 317.30 | 341.70 | 361.10 | 377.10 | 392.80 | 409.80 | 433.20 |
| 160 | 247.00 | 290.18 | 329.40 | 363.10 | 390.20 | 412.20 | 430.40 | 447.80 | 466.60 | 492.80 |
| 170 | 283.10 | 331.70 | 375.60 | 413.30 | 443.70 | 467.80 | 488.70 | 508.20 | 528.80 | 558.40 |
| 180 | 323.10 | 377.10 | 426.60 | 468.40 | 502.40 | 529.50 | 552.30 | | | |
| 190 | 367.10 | 427.70 | 482.50 | 528.80 | | | | | | |
| 200 | 415.10 | 483.00 | 543.60 | | | | | | | |
| 210 | 468.40 | 542.90 | | | | | | | | |
| 220 | 525.50 | | | | | | | | | |
| 230 | | | | | | | | | | |
| 240 | | | | | | | | | | |
| 250 | | | | | | | | | | |

Figures 55 to 58 show details of the absorption machine, and Table 16 gives the surfaces used in standard machines. Also Table 18 gives the total vapor pressure of aqua ammonia and Fig. 60 the partial pressures

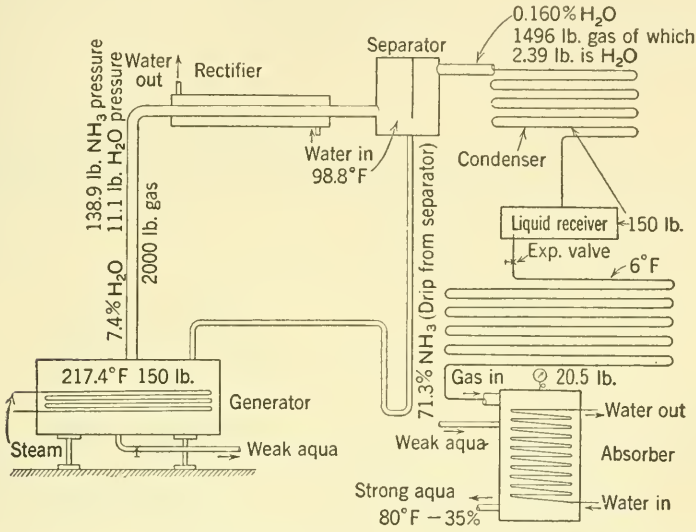



Fig. 61.—Schematic Arrangement of the Absorption Machine.

of water vapor above aqua ammonia. These values and the chart are taken from Bulletin No. 146 of the Engineering Experiment Station of the University of Illinois.

**The Absorber.**—The action of the absorber is a complex one, not easily defined in a single statement. The gas from the evaporator enters the absorber and comes in contact with a spray of weak, cooled aqua, and a strong aqua is formed. For simplicity the process in the absorber is divided into three parts, namely, *a*, the cooling of the weak aqua to the temperature of the strong aqua leaving the absorber, *b*, the condensation of the gas from the evaporator, and *c*, the absorption of the liquid ammonia by the weak aqua. The pressure in the evaporator is determined by that in the absorber which in turn is a function of the temperature and concentration of the strong aqua in the absorber. The process of liquefaction and of solution is therefore at constant pressure,

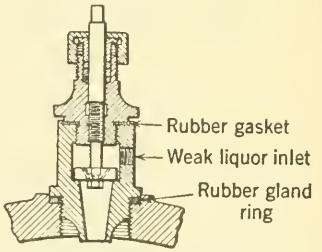



Fig. 62.—Mixing Device in Absorber.

but the temperature of liquefaction (the temperature of anhydrous ammonia corresponding to the pressure) is much less than that of the strong aqua. In consequence the liquid ammonia formed by the liquefaction previous to the entering into solution must be conceived to be *heated* up to this temperature before solution takes place.

In order to bring out the method of calculation the following problem will be solved:

**Problem.**—The strong aqua leaving the absorber has a concentration of 35 per cent, and a temperature of 80 deg. F. The weak aqua has a concentration of 30 per cent and enters the absorber at a temperature of 120 deg. F. The heat absorbed by the cooling water is required and the number of pounds of water per minute, assuming a temperature rise of 10 deg. F.

The number of pounds of *weak* aqua circulated (Table 15) in order to absorb 1.0 lb. of ammonia gas is given by the formula,

$$n = \frac{1 - x_2}{x_1 - x_2} - 1.0 = \frac{1.0 - 0.30}{0.35 - 0.30} - 1.0 = 13.0 \text{ lb.}$$

where $x_1$ is the concentration of the strong, and $x_2$ that of the weak aqua.

The pressure in the absorber, at 35 per cent concentrations and 80 deg. F. is 20.5 lb. The weak aqua is 30 per cent ammonia and 70 per cent water. It has been the practice to assume that the specific heat of the aqua is found by assuming that simply a mixture existed. The specific heat of the liquid ammonia in this case is 1.175, therefore the specific heat of the mixture is

$$(0.30 \times 1.17) + (0.70 \times 1.0) = 1.053$$

*a.* The cooling effect necessary to cool the weak aqua per 1.0 of ammonia.

$$Q = Mc(t_2 - t_1) = 13.0 \times 1.053 \times (120 - 80) = 547.6 \text{ B.t.u.}$$

*b.* Cooling required for the liquefaction, etc.

$$Q = 606.5 - 132.0 = 474.5 \text{ B.t.u.}$$

*c.* Cooling required to overcome the heat of solution. This is expressed by

$$Q = 345(1 - x) - 400x^2 \text{ where } x \text{ is the average concentration}$$
$$= 345 \times 0.675 - 400 \times (0.325)^2 = 190.8 \text{ B.t.u.}$$
$$\text{Total} = 1212.9 \text{ B.t.u.}$$

The amount of water required with a 10-degree rise of temperature will be

$$1212.9 \div 10 = 121.3 \text{ lb.} = 14.56 \text{ gallons}$$

If a standard condenser pressure was carried in the condenser the ammonia required per ton of refrigeration would be—at 20.5 lb. and 86 deg. F. condensing temperature

$$200 \div (606.5 - 138.9) = 0.428 \text{ lb.}$$

and the water per ton per minute required in the absorber would be

$$14.56 \times 0.428 = 6.23 \text{ gallons}$$

**The Munters-Platen Absorption System.**—The Munters-Platen system, shown in Fig. 63, is an attempt to eliminate the strong aqua pump. The pressure is exerted throughout the entire system and no "expansion" valve is required. Circulation is obtained by the action of the heating element in the generator. The outstanding feature of the system is the use of an inert gas (hydrogen) in the evaporating coil; sufficient so that the *partial pressure* of the ammonia will be that required by the conditions of operation.

While this system is ingenious it requires several constant conditions for success. The total pressure must remain constant[6] and there must be no *leakage* of hydrogen. As most refrigerating machines have a vari-

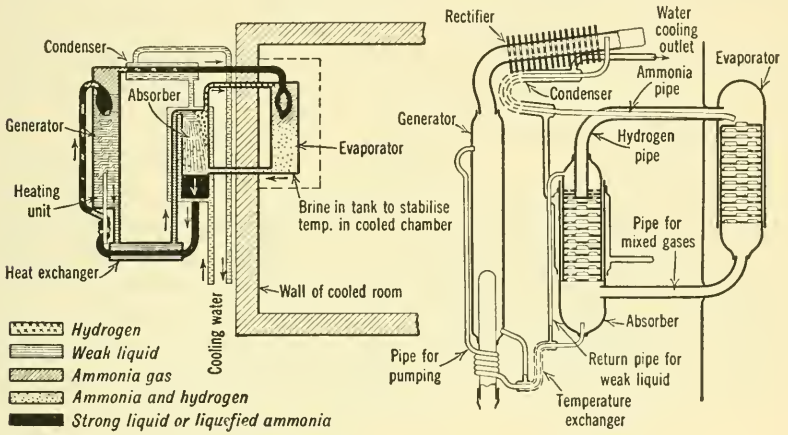



FIG. 63.—The Munters-Platen System.

able condenser pressure depending on the amount and the temperature of the condensing water it is not clear how successful the Munters-Platen system will be. It undoubtedly will be limited, under any circumstances, to the gas-fired machine.[7]

[6] The pressure in the system may be taken as $p$ lb. per square inch, which is equal to $p_1 + p_2$ in the evaporator coils where:

$p_1$ = the pressure exerted by the ammonia
$p_2$ = the pressure exerted by the hydrogen

Suppose the condenser pressure $p$ increases, due to a temperature rise of the condensing water. Then $p_1$ will increase a like amount as the change of pressure of the hydrogen would be small. The temperature of evaporation would be regulated then by the condenser pressure, a condition which can be controlled to some extent, but usually is not, except to make the condenser pressure as low as possible.

[7] A modification of the Munters-Platen system is the Altenkirch refrigerating machine. This device uses sulphuric acid as a solvent (La Technique Moderne, July 1, 1926).

**The Intermittent Absorption Machine.**—The intermittent absorption machine is of particular interest for the small sizes, as the chief feature of the design is the elimination of the strong aqua pump and therefore the necessity for the use of power in the machine. It is automatic as regards the water control and the heating element, which is gas fired always because of the cost, and a part of the machine acts both as a generator and an absorber. Quite naturally refrigeration cannot be continuous. However, several large installations have been made, as for example in milk cooling after pasteurization.

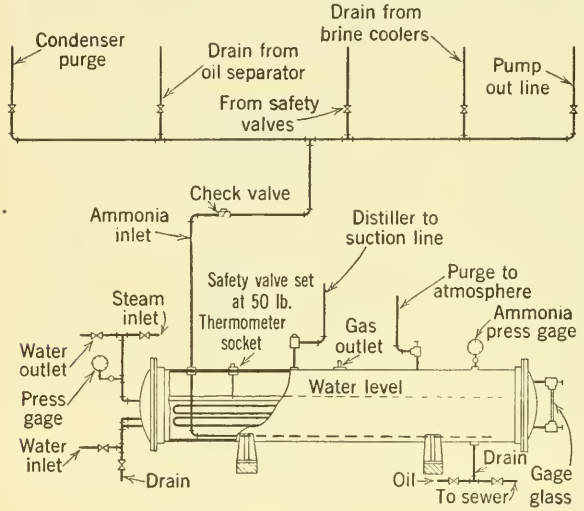



Fig. 64.—Purge Drum for Absorption Machine.

One of the essential features of the absorber is the thorough mixing of the gas and weak aqua. This is accomplished in Fig. 62 by the device of the weak aqua nozzle which forms a spray and permits the aqua in this condition to meet the ammonia gas from the evaporating coils. Another mixer is the double pipe with the inner pipe perforated. The gas passing through the holes in the inner pipe as a series of bubbles meets the liquid in the outer pipe.

**Purging.**—Figure 64 is a special device adapted to large plants to remove the non-condensible gases. Some disintegration occurs in the absorption machine and air enters in one manner or another, so that purging of the absorber is necessary. Purging always means a loss of ammonia unless a special device is used which in this case is a special absorber generator. As an absorber, water cooled, the ammonia in the purge gas is absorbed and the inert gas is blown off. As a generator the steam coils drive off the ammonia in the still where it is discharged into the condenser.

## CHAPTER IV

## FITTINGS AND CONDENSERS

**Ammonia Fittings.**—Because of the nature of ammonia it is necessary to make especial provision against leaks, even though the pressures are not more than 200 lb. per square inch as a rule. The accepted form of flanged joint is that kind employing the tongue and groove, using generally a rubber or lead gasket, although asbestos and other material are sometimes used. Fittings may be designed with screw ends—up to 3 to 5 in.,—with oval, square and round flanges, depending on the size of the pipe.

There are two methods of making up pipe and fittings,[1] the first method being the litharge and glycerine, and the second being the solder joint in the case of black iron and steel pipe. As a rule the latter is confined to condensers and (possibly) to brine coolers that are made up at the factory, and where the necessary equipment is at hand for the purpose. The process is to tin the pipe—using a bright pipe and screw threads and an acid composed of muriatic acid that has been partly neutralized by the action of zinc—and then to make up the joint with the pipe and the fitting heated up to the temperature of the solder. In the larger sizes of fittings there are solder recesses (Fig. 65) to be filled with solder and this is caulked in order to cover the threads. According to Fairbanks[1a] the defects of soldering are two in number. First, in the larger pipes—3 in. and over—vibration, expansion and contraction of the pipe finally break the soldered joint with the result that a leak develops. Second, the temperature of the discharge connections from the compressor is often 300 deg. F. in which

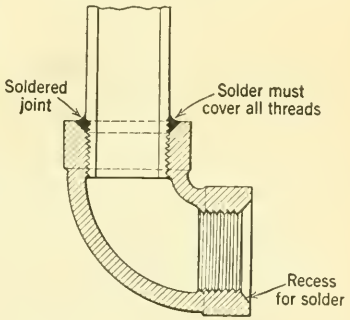



Fig. 65.—Fitting with Solder Recess.

[1] For a more complete discussion of erection problems see Chapter X.

[1a] Fairbanks in the Mechanical Engineers Handbook, McGraw-Hill Book Company, p. 1729.

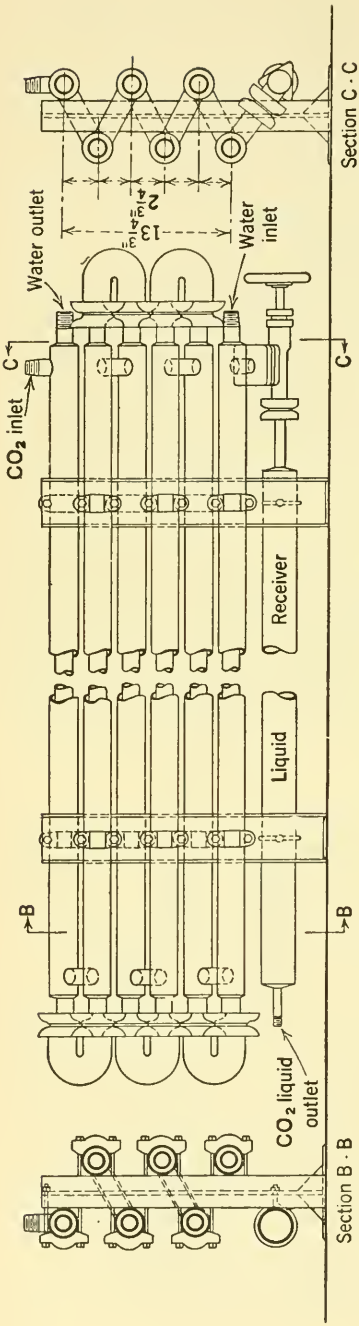



FIG. 66.—Double Pipe Welded $CO_2$ Condenser.

case the solder becomes plastic. Fairbanks favors flanging and V grooving the flange for all pipe 3 in. and over, in which case full-weight pipe only is used, employing the Van Stone joint and an antimonial lead gasket.

The essential feature of the ammonia pipe connection is a clean and accurate thread. The amount of pull or the heat developed during the making up of the threaded joint is not a reliable evidence of a good pipe connection as these factors could be caused by dirt. In order to secure good work the threads must be cleaned carefully, preferably by means of gasoline, and assurance obtained that chips, and other loose pieces of steel, are not caught in the threads. It is usual to pull up on the pipe threads much more than for similar work in steam, and the result is that all screw fittings are much heavier than those designed for water or steam for 250 lb., and in addition the cast iron of the fittings must be of a dense structure and always heavier-walled than a steam fitting would be. For some time the close-grained, air furnace, gray iron casting was used but this is being replaced by the semi-steel and the drop-forged steel fitting on the part of some of the manufacturers. These fittings preferably should be malleable for screwed and semi-steel for flanged fittings and drop forged for flanges. The drop-forged steel fitting is becoming more popular, although more expensive, be-

cause of its lack of porosity, its great strength and the fact that it can be welded to the pipe. This last property is not so very important with ammonia machines but very desirable in connection with carbon dioxide refrigeration.

In ammonia work it has been found best to use a soft gasket, and to form the joint by a method which will prevent the gasket from squeezing out. The tongue and groove design absolutely prevents the blowing out of the gasket material, and the soft lead asbestos gasket seems to give good service under these conditions. The ordinary copper and asbestos gasket—with the flat-faced flange—is never attempted, although the Fairbanks design is somewhat like the flat-faced flange, both on account of inability to secure a sufficient unit pressure on the packing and also because the asbestos will flatten and the copper will corrode (if water is present in the ammonia) and flatten when under ammonia pressure. The present tendency is to eliminate valves and fittings as far as possible by using flame, electric and forge welding at the shop and flame and thermit welding on the job. Pipe work can be made welded entirely except for the unions and perhaps the stop and the expansion valves. Some double pipe condensers are made—especially for carbon dioxide or for small capacity plants—welded throughout (Fig. 66); and apparently welding will supersede fittings whenever the cost as well as the conservation of the refrigerant makes this justifiable. For evaporator pipe coils the flanged bent pipe is rapidly replacing the cast return bend (Fig. 67) except when the pipes are submerged in a liquid, in which case continuous welded pipe is used. It cannot be said, however, that there is any great tendency to do away with the double pipe and the atmospheric type of condenser for ammonia, both of which are plentifully supplied with fittings and valves.

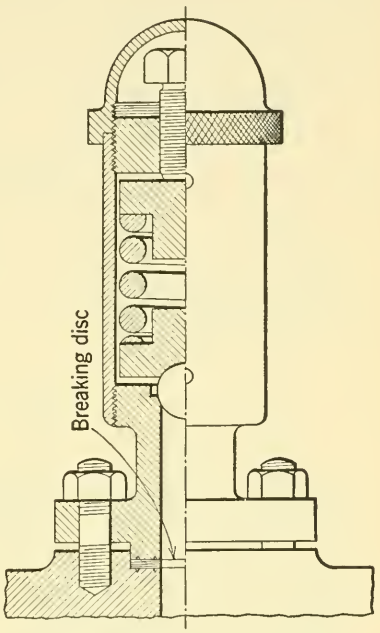



FIG. 67.—Safety Valve for Carbonic Compressors

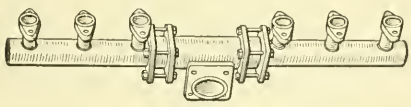


FIG. 68.—Welded Headers.

**Manifolds and Headers.**—Manifolds and headers are made either of semi-steel castings or are made up of steel and welded, for which purpose the autogenous (flame) weld is very successful (Fig. 68). This

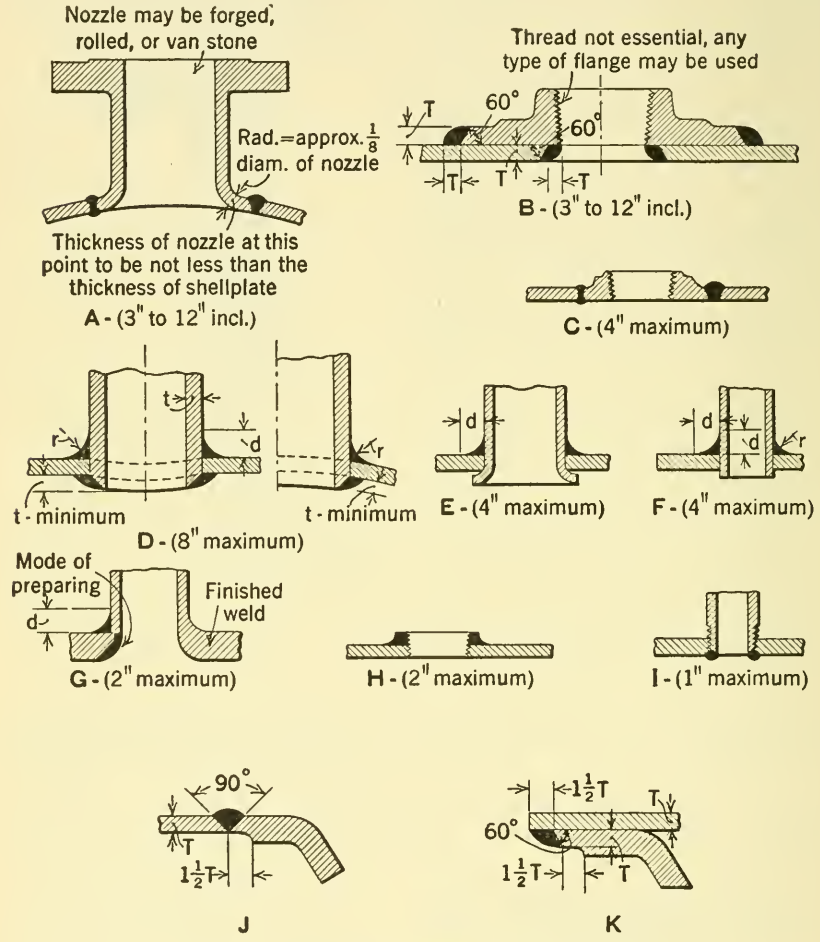

Nozzle may be forged, rolled, or van stone
Rad.=approx. 1/8 diam. of nozzle
Thickness of nozzle at this point to be not less than the thickness of shellplate
A - (3" to 12" incl.)
Thread not essential, any type of flange may be used
60°
T
B - (3" to 12" incl.)
C - (4" maximum)
t - minimum
D - (8" maximum)
E - (4" maximum)
F - (4" maximum)
Mode of preparing
Finished weld
G - (2" maximum)
H - (2" maximum)
I - (1" maximum)
90°
J
K
Note:- For all nozzles, dimensions d and r shall not be less than the following:
d = Nominal pipe size / 8 + 1/4"
r = Nominal pipe size / 2

Fig. 69.—Methods of Flame Welding.

is the method whereby the gases in two tanks, one of acetylene ($C_2H_2$) and the other of oxygen, are mixed in a nozzle and the mixture is controlled by separate valves. The weld is made by fusing the material

to be welded and by melting a steel rod until the proper reinforcement is obtained (Fig. 69).[2] The flame weld has not been entirely successful, and for certain work the forge weld has taken its place, as for example in the welded heads of liquid receivers. Also for the same reason the flame weld along the longitudinal seam is being done away with, wherever convenient, by the use of the proper sized lap welded pipe manufactured in the steel mill, and pipe up to 90 in. diameter is obtainable.

**Valves.**—Valves for ammonia are usually of the globe and the angle type. When used as stop valves they may be furnished with either

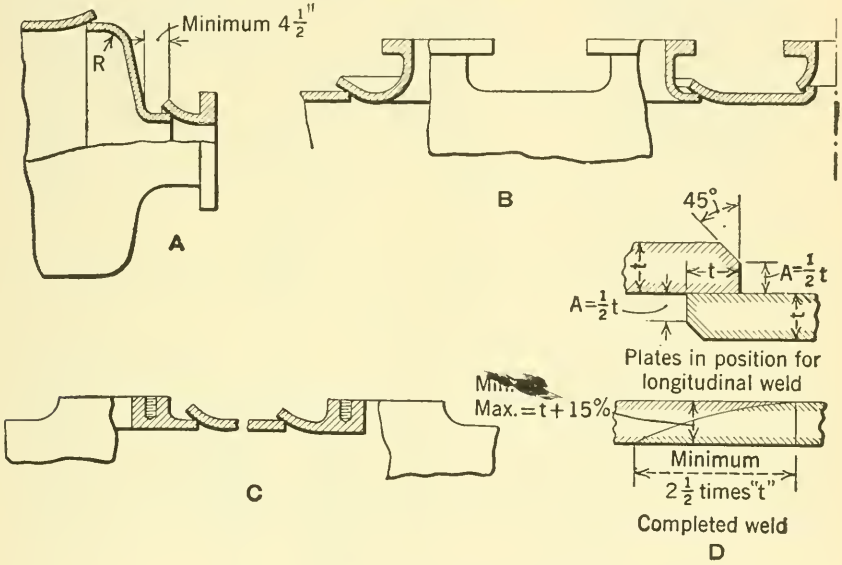



Fig. 70.—Methods of Flame Welding.

hard or soft seats. The advantage of the soft, white metal seat—dovetailed into the valve disc—is that foreign material, like steel cuttings, will sink into the soft material instead of preventing the valve from seating and therefore causing a leak, but the soft seat has to be renewed at regular intervals. Usually the valve is so designed as to make a seat at the *top* of the valve when it is full opened (Fig. 71) in order that leakage past the stem through the packing when the valve is open may be prevented and also so that the valve stem may be repacked while the valve is under pressure.

Special designs of regulating (expansion) valves are numerous. These have a special fine thread on the spindle for fine adjustments,

[2] Rules for the construction of unfired pressure vessels, A. S. M. E. Boiler Construction Code.

and are made with either a conical projection of the spindle through the valve seat, or a cylindrical projection with a tapered groove in order to obtain the required close adjustment. When closed the expansion valve (Fig. 71) can be made tight by means of a soft metal seat ring or by means of a plain needle-type valve of hard steel. When such close adjustments are not required a modified globe or angle valve may be used.

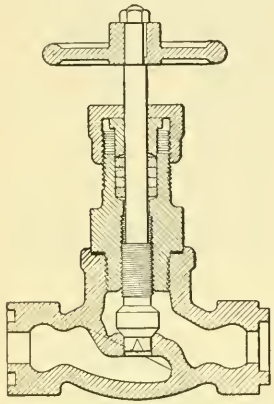


Fig. 71.—Section of an Expansion Valve.

The expansion cock is still used but it is inclined to jam, and at the present time it is not very popular. The flow of one cubic foot of liquid ammonia per minute is sufficient for approximately 100 tons of refrigeration, and with a difference of pressure of 150 lb. per square inch on the two sides of the valve a large opening is not required. On the other hand with the use of the expansion cock the suction pressure is harder to regulate than with the usual expansion valve.

Figures 72 to 85 show various forms of fittings. Figure 80 is one design of strainer and Fig. 81 one form of an ammonia purifier. Figures 83, 84 and 85 and Table 19 give dimensions of some of the standard large accessories. An isometric sketch of the low-pressure suction return to the compressor, showing pipes, fittings and an accumulator is given in Fig. 86. Tables 22 to 31 give dimensions of the more common

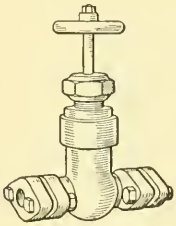


Fig. 72.—Expansion Valve.

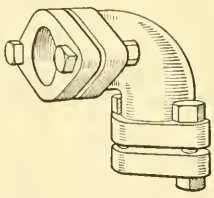


Fig. 73.—Elbow.

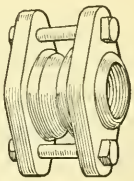


Fig. 74.—Flange.

York fitting, Tables 32 to 35, inclusive, give Frick practice, Tables 36 and 37 show Arctic, whereas Tables 20 and 21 give the dimensions of Crane flanges. It will be noticed on comparing these tables that even the flanges for ammonia are not standard either as regards the diameter, the width or the depth of the groove, and even the thickness of the metal, size and spacing of the bolts, the distance between faces, etc.

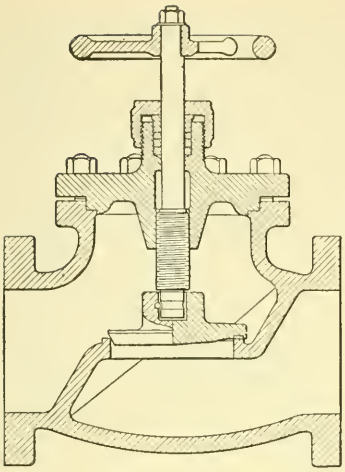


Fig. 75.—Section of Globe Valve.

bear no resemblance to one another in the different makes. Very unfortunately there does not appear at present any possibility of standardization. The Crane fitting dimensions for ammonia are taken from what is known as "extra heavy steam," and the metal thicknesses,

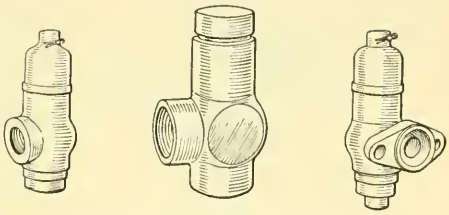


Fig. 76.—Pop Safety Valves.

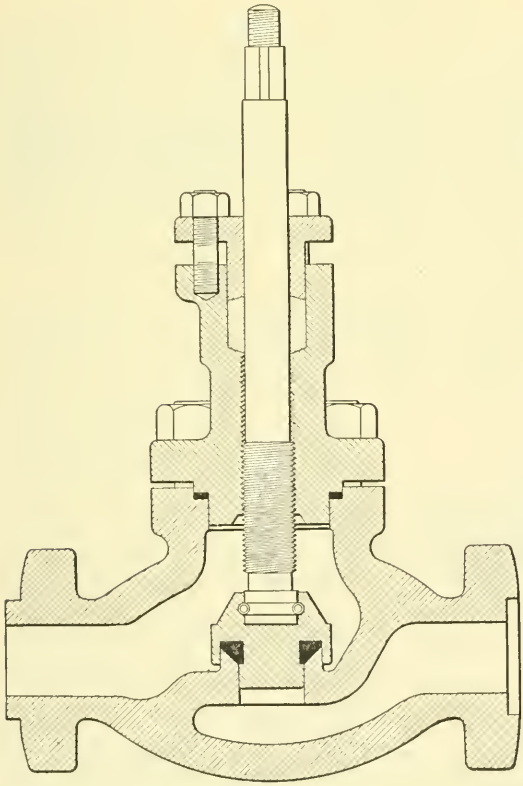


Fig. 77.—Section of Globe Valve for $CO_2$.

the size of flanges and boltings and the size of the bolts are the same as for their steam fittings and valves.

**Carbon Dioxide Fittings.**—Figures 87 and 88 show typical carbonic fittings. Figure 88 shows a flange joint for a large pipe wherein the gas seal is made by means of lead gaskets made tight by means of an annular gland. In the smaller sized pipe flanges the joint has been made by permitting the pipe (the ends of which have been reamed square) to project $\frac{1}{32}$ in. beyond the flange, and a hard red fiber gasket has been used to make up the joint. However the hard fiber has a

tendency to crack and later practice has been in the direction of a composition aluminum or copper gasket. Figure 88 shows a typical $\frac{1}{2}$ by $1\frac{1}{4}$ expansion valve which is noteworthy only on account of the heavy metal construction.

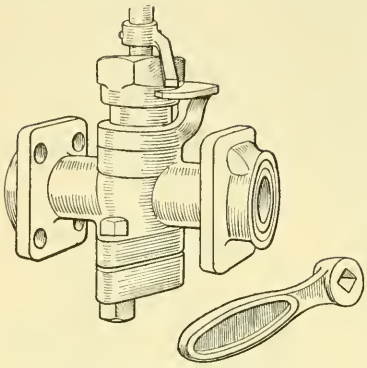


Fig. 78.—Expansion Cocks.

Carbonic fittings are not standard and each manufactory has its own patterns. It is becoming more and more common to make use of drop-forged fittings for carbon dioxide or, more generally speaking, for the line of fittings designed for 3000 lb.

## CONDENSERS

**Historical.**—In the early days of refrigeration the condenser was nothing else than a round or a flat coil of pipe made up with suitable fittings at first and then, as time elapsed, the pipe was welded completely or was made up with a combination of welding and fittings. The refrigerating engineer was obsessed with the idea of the danger due to ammonia and the difficulty in reducing ammonia leaks, the latter being important in the last quarter of the nineteenth century because of the heavy cost. The refrigerating engineer felt that he could not take chances of rupture, but he knew that the pipe condenser could be made safe and reasonably tight. The submerged and the atmospheric condenser, made up with pipe and fittings, were the result, and later the shell and coil condenser was developed.

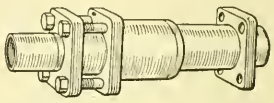


Fig. 79.—Expansion Joint for Ammonia.

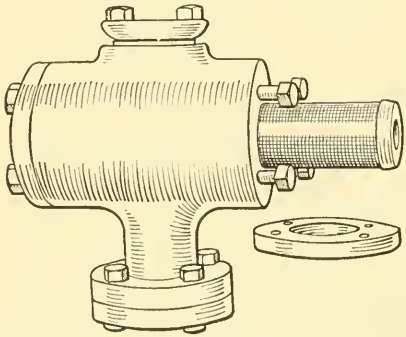


Fig. 80.—Strainer.

As time elapsed, and with the resulting experience and improved materials available for use, the refrigerating engineer devised more complicated fittings which resulted in the double-pipe counterflow condenser and later still this was modified for carbon dioxide refrigera-

tion by welding throughout. Finally came the shell and tube condenser, similar in construction to the steam surface condenser, but with the tubes rolled into both of the tube sheets. At first the shell was

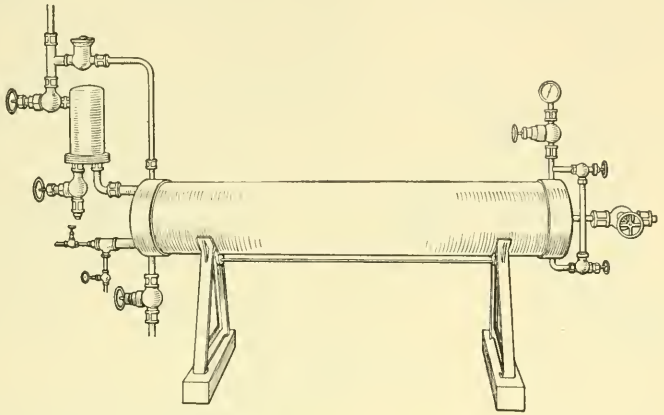


FIG. 81.—Ammonia Distiller.

riveted as well as welded, but the practice in 1926 was usually to take the lap-welded pipe as manufactured at the steel mills. This condenser was designed for vertical and horizontal shells, but the vertical is preferred and very nearly the only one used in the larger sizes.

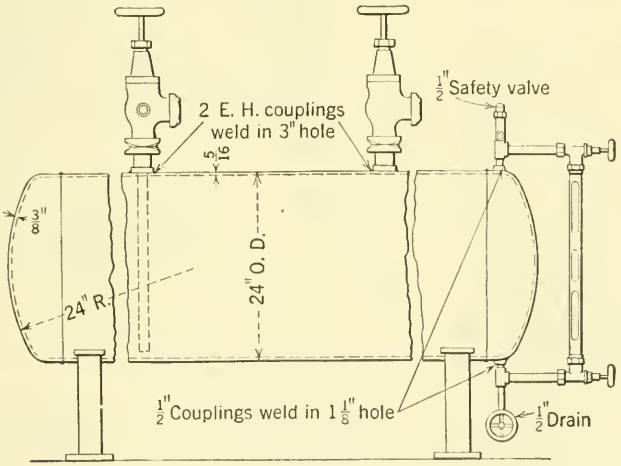



FIG. 82.—Ammonia Liquid Receiver.

As compared with steam engineering it would seem that the progress in the development of the ammonia condenser has been very slow. The

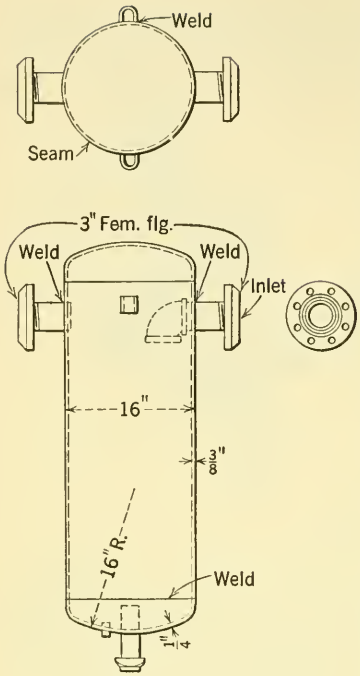



Fig. 83.—Accumulator.

made without great difficulty.

**Function of the Condenser.**—The condenser is for the purpose of removing the superheat and the latent heat of liquefaction, and except in very special cases, it is water cooled. The temperature of the gas entering the condenser may be as high as 250 deg. F. or even 300 deg. F. and the amount of heat removed varies with the operating conditions from 220 to 260 B.t.u.

pipe condenser appears to be too complicated, and to be unnecessarily supplied with valves and fittings. In addition the space occupied is excessive at times, and certain of the designs are difficult to arrange so that the different stands will have an equal amount of the load. It seems at first as if the design of condenser made up of pipes and fittings must be inefficient when one considers that it is, in all intents and purposes, a long pipe which may have a dead end or is connected to a liquid receiver, a design which has hardly a counterpart in steam engineering.

On the other hand the pipe designs of condensers have given good service, are reasonably low in first cost and have been economical in the use of water. They can be erected so that the leakage loss is very small and repairs (if such are required) can be

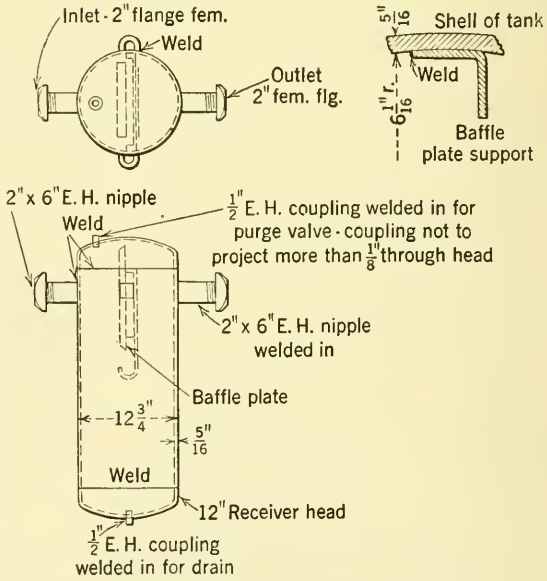



Fig. 84.—Oil Separator.

per ton per minute. Some designs function partly as liquid receivers, particularly in the case of carbon dioxide, but this is a minor object of the condenser and one which is being gradually discarded as a characteristic of the apparatus. The main object is

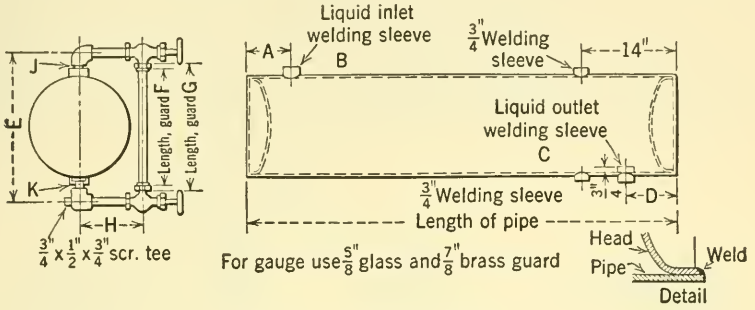



FIG. 85.—Liquid Receiver.

| Diameter of Pipe | Length of Pipe | | Contents, Cubic Feet | Thickness | | A | B | C | D | E | F | G | H | J | K |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Of pipe | Of heads | | | | | | | | | | |
| | Ft. | In. | | | | | | | | | | | | | |
| 10 | 8 | 0 | 4.4 | | | | | | | | | | | | |
| 10 | 10 | 0 | 5.5 | | | | | | | | | | | | |
| 10 | 14 | 0 | 7.7 | 3/8 | 1/2 | 8 | 1 1/4 | 1 | 8 | 17 | 11 3/4 | 14 | 6 1/2 | 1 3/8 | 2 |
| 10 | 18 | 0 | 9.9 | | | | | | | | | | | | |
| 12 | 15 | 0 | 11.8 | | | | | | | | | | | | |
| 12 | 17 | 6 | 13.8 | 3/8 | 1/2 | 8 | 1 1/2 | 1 1/4 | 8 | 19 | 13 3/4 | 16 | 7 1/2 | 1 3/8 | 1 3/8 |
| 12 | 20 | 0 | 15.7 | | | | | | | | | | | | |
| 16 | 15 | 0 | 19.0 | | | | | | | | | | | | |
| 16 | 17 | 6 | 22.2 | 3/8 | 5/8 | 8 | 2 | 2 | 8 | 23 | 17 3/4 | 20 | 9 1/4 | 1 3/8 | 2 |
| 16 | 20 | 0 | 25.4 | | | | | | | | | | | | |
| 20 | 14 | 0 | 27.9 | | | | | | | | | | | | |
| 20 | 17 | 0 | 33.9 | 1/2 | 1 | 8 | 2 | 2 | 8 | 27 | 21 3/4 | 24 | 11 1/4 | 2 | 2 |
| 20 | 20 | 0 | 40.0 | | | | | | | | | | | | |
| 24 | 16 | 0 | 46.2 | | | | | | | | | | | | |
| 24 | 20 | 0 | 57.7 | 9/16 | 1 1/8 | 8 | 2 | 2 | 8 | 31 | 25 3/4 | 28 | 13 1/4 | | |

All dimensions given in inches unless otherwise specified.

to liquefy the gas at as low a temperature and pressure as is possible with the quantity and temperature of cooling water available at the plant. How this pressure varies with different conditions of cooling water is shown in Table 75.

**Types of Condensers:** *The Submerged Condenser.*—The submerged condenser is an obsolete design for ammonia at the present time except

for marine work and carbonic refrigeration. It consists of a coil or a set of concentric coils placed within the shell or tank. The refrigerant passes through the coils and the water is within the shell or tank outside the pipe coils. The coils must be made up without joints below the water surface and they are preferably welded. When more than one coil is used they are brought together in a header. With carbon dioxide the submerged condenser is still used to some extent, and the design can be made more efficient in heat transfer by the use of special devices to increase the velocity of the water flow past the surface of the pipe

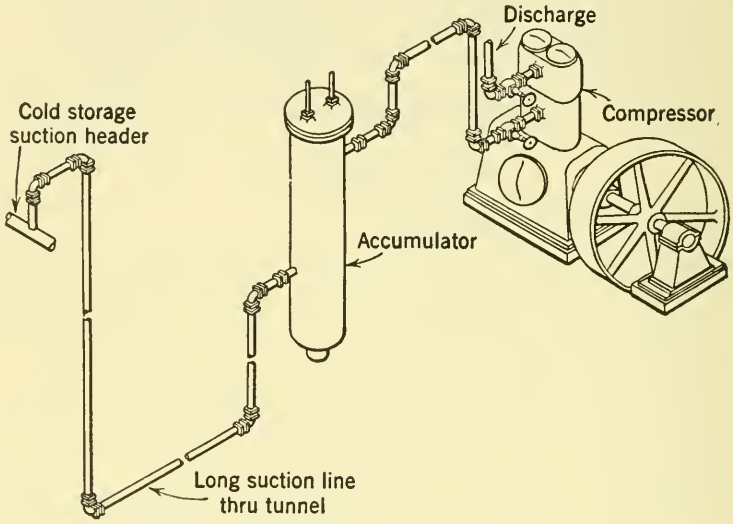
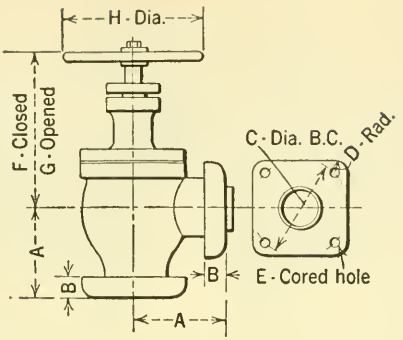



Fig. 86.—Method of Piping an Accumulator.

coils by the use of an inner pipe or by the use of suitable baffles (Fig. 89). In the old designs the rate of heat transfer between the refrigerant and the water is approximately 0.2 of that in the better types of condensers, but the later designs for carbon dioxide have improved the performance considerably, although there is difficulty in cleaning the coils if dirty or scale forming water is used.

*The Atmospheric Condenser.*—The atmospheric condenser is the general term for that design which is made up in stands with the water delivered to the outside of the upper pipe with the succeeding pipes receiving the water by dripping from one pipe to the next. The gas from the compressor may enter at the top (the old type), at the bottom for about three pipes and then pass to the top (the common atmospheric condenser) or at the bottom (the bleeder type). The British design of atmospheric condenser consists frequently of sheets of pipes, usually

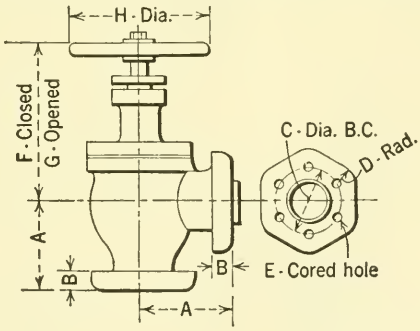



*a*

TRISEAL ANGLE VALVE
4-Bolt Type

| Size | Symbol Number | A | B | C | D | E | F | G | H | Number of Bolts | Size of Bolts |
|---|---|---|---|---|---|---|---|---|---|---|---|
| $\frac{3}{4}$ | 1206-F | $3\frac{1}{4}$ | $\frac{7}{8}$ | $3\frac{1}{8}$ | $\frac{5}{8}$ | $\frac{5}{8}$ | $9\frac{1}{2}$ | $10\frac{1}{4}$ | 5 | 8 | $\frac{1}{2}\times2\frac{3}{4}$ |
| 1 | 1202-F | $4\frac{3}{16}$ | $1\frac{1}{16}$ | $3\frac{1}{2}$ | $\frac{3}{4}$ | $\frac{3}{4}$ | $11\frac{1}{2}$ | $12\frac{1}{2}$ | 7 | 8 | $\frac{5}{8}\times3\frac{1}{4}$ |
| $1\frac{1}{4}$ | 1180-F | $4\frac{5}{16}$ | $1\frac{1}{8}$ | $4\frac{1}{8}$ | $\frac{3}{4}$ | $\frac{3}{4}$ | 11 | $12\frac{3}{4}$ | 7 | 8 | $\frac{5}{8}\times3\frac{1}{4}$ |
| $1\frac{1}{2}$ | 1240-F | $5\frac{3}{16}$ | $1\frac{1}{8}$ | $4\frac{1}{2}$ | $\frac{3}{4}$ | $\frac{3}{4}$ | 14 | $15\frac{1}{2}$ | $9\frac{1}{2}$ | 8 | $\frac{5}{8}\times3\frac{1}{4}$ |
| 2 | 1232-F | $6\frac{1}{8}$ | $1\frac{1}{4}$ | $5\frac{1}{8}$ | $\frac{7}{8}$ | $\frac{7}{8}$ | $13\frac{1}{2}$ | $15\frac{1}{2}$ | $10\frac{1}{2}$ | 8 | $\frac{3}{4}\times3\frac{3}{4}$ |

All dimensions given in inches unless otherwise specified.




*b*

TRISEAL ANGLE VALVE
6-Bolt Type

| Size | Symbol Number | A | B | C | D | E | F | G | H | Number of Bolts | Size of Bolts |
|---|---|---|---|---|---|---|---|---|---|---|---|
| $2\frac{1}{2}$ | 1284-F | $6\frac{13}{16}$ | $1\frac{5}{8}$ | $6\frac{1}{2}$ | 1 | 1 | $14\frac{1}{2}$ | $16\frac{1}{2}$ | $10\frac{1}{2}$ | 12 | $\frac{3}{4}\times4\frac{1}{2}$ |
| 3 | 1286-F | $7\frac{5}{16}$ | $1\frac{3}{4}$ | $7\frac{3}{4}$ | 1 | $1\frac{1}{8}$ | $17\frac{3}{4}$ | $20\frac{1}{2}$ | 13 | 12 | $\frac{7}{8}\times5$ |
| 4 | 1320-F | $10\frac{3}{4}$ | 2 | $9\frac{3}{4}$ | $1\frac{1}{4}$ | $1\frac{1}{4}$ | $21\frac{1}{2}$ | $24\frac{5}{8}$ | 13 | 12 | $1\times5\frac{1}{2}$ |

All dimensions given in inches unless otherwise specified.

FIG. 87.—Carbonic Fittings, Angle Valves.

three in number, interlaced so as to be vertically over one another. In this design, called the evaporative open condenser, the number of pipes may be as many as 48 high; and the water, being supposed to be cooled by the air and by evaporation, is recirculated. In America the atmospheric condenser which was at first 30 pipes high has been reduced to 18 and finally to 12 except in certain particular designs and no attempt at recirculation of the water is made unless a separate system of water cooling is provided by means of a cooling tower or by the use of sprays.

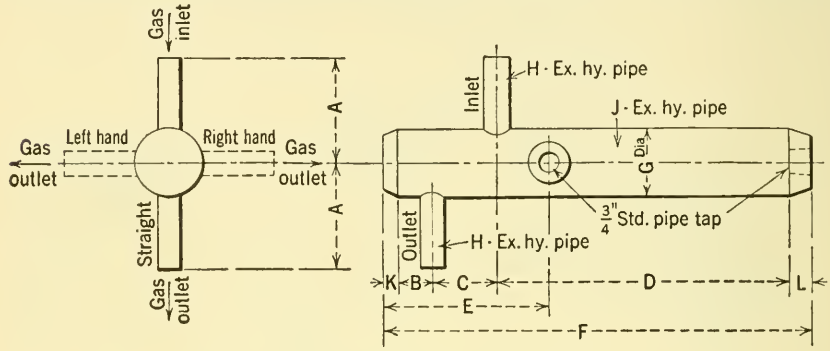



Fig. 88.—Carbonic Fittings, Oil Separator.

Welded Type Oil and Scale Traps

| Size | Symbol Number | A | B | C | D | E | F | G | H | J | K | L |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| ¾ | 1321-K | 4½ | 1½ | 2½ | 12½ | 7 | 18 | 2⅞ | ¾ | 2½ | ⅝ | ⅞ |
| 1 | 1321-H | 4½ | 1¾ | 2¾ | 16 | 9 | 22 | 2⅞ | 1 | 2½ | ⅝ | ⅞ |
| 1¼ | 1321-F | 5 | 2¼ | 3¼ | 18½ | 9 | 25½ | 3½ | 1¼ | 3 | ⅝ | ⅞ |
| 1½ | 1321-D | 5¼ | 2¼ | 3¼ | 18½ | 9 | 25½ | 3½ | 1½ | 3 | ⅝ | ⅞ |

All dimensions given in inches unless otherwise specified.

The design of atmospheric condensers has created considerable interest for a number of years. First came the Block and the Shipley designs of " flooded " condensers whereby the gas from the compressor was mixed with liquid ammonia by means of a sort of ejector nozzle and during the remainder of the process the mixture of gas and liquid was cooled by metal contact with the condensing water as in the other designs of atmospheric condensers. The theory advanced was that the superheat in the ammonia decreased the rate of heat transfer considerably, and that the superheat could be removed quickly by means of this liquid injection. Tests by Ophuls and Greene[3] disproved some of these contentions. Louis Block in the 90's devised a counterflow atmospheric

[3] Ophuls and Greene, American Society of Refrigerating Eng. Journal, 1916.

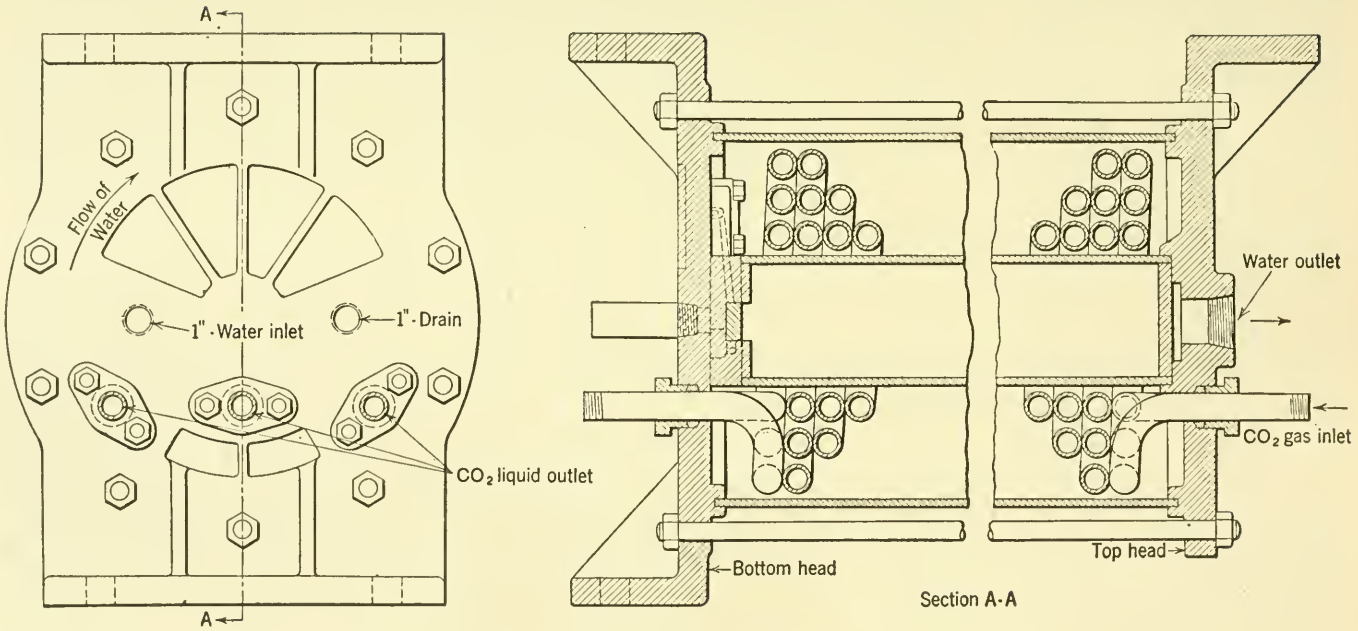



FIG. 89.—Submerged Type $CO_2$ Condenser.

condenser, bringing in the gas at the bottom and making the upper pipe a dead end. In order to remove the liquid he drained off the condensate at frequent intervals and his condenser became known as the *drip pipe* or *bleeder* condenser. Block's whole design hinged about the importance of securing a counterflow of the refrigerant and the water, an arrangement which had been considered essential for a long time, but which was actually accomplished in the Block (the De la Vergne) design by the use of a large number of fittings. However, recent experiments are changing the opinion as to these condensers, and it is doubtful whether the bleeder or the flooded condenser will be very popular in the future.

Tests in the Engineering Experiment Station of the University of Illinois on a bleeder type of condenser[4] using two thermocouples per pipe for the ammonia temperatures and a carefully designed measuring device for the condensate indicated that from 40 to 70 per cent of all the liquefaction occurred in the upper pair of pipes, the larger amount of condensate occurring at the time that the coldest water was used. During these tests the condenser pressure was constant, and variation was possible by the use of different initial temperatures of the cooling water, and the amount of water used was that required to maintain a constant head pressure. In other words approximately 50 per cent of the gas traveled through some 200 ft. of condenser pipe before liquefaction occurred. Also there was evidence that at certain times the upper pipes filled with condensate so that the load was carried by the next lower pipe. The superheat was found to be removed in the second or third pipe, depending on the suction pressure, which varied the load on the condenser, whereas the fourth, fifth, sixth and the seventh pipe from the bottom carried no appreciable load as they were brought into metal contact with water very near the temperature of liquefaction. The temperature of the ammonia in the condenser was practically constant in the upper nine pipes.

Similar tests on a double pipe condenser[5] indicated very nearly similar conditions to those of the bleeder condenser, except that the maximum liquefaction takes place in the lowest pair of pipes, which are in this case cooled with the coldest water. Here again the temperature of the ammonia was found to be practically constant in the lowest nine pipes. In this case also approximately 50 per cent of the ammonia gas traveled through nearly the entire condenser before liquefaction occurred, and the temperature of the gas during condensation remained constant as theoretical considerations would lead one to expect.

[4] This condenser consisted of 12 2-in pipes, approximately 20 ft. long.

[5] This was $1\frac{1}{4} \times 2$, 12 pipes high, approximately 20 ft. long.

These tests would seem to point out certain fallacies. If the temperature of the ammonia remains constant throughout the condenser where liquefaction occurs there is no material advantage in having a counterflow as the mean temperature between the water and the ammonia will be identical in either case. However, there are designs which utilize the surface more effectively than others. If the superheat is removed in two or three pipes there is no advantage in making the condenser more complex in order to get rid of the superheat. In other words, that part of the condenser exposed to saturated ammonia has drops of liquid forming on the surface, and in consequence the surfaces are wet. In general it has been demonstrated by tests that heat transfer is decidedly affected by the condition of the surface. A layer of inert refrigerant, air or decomposed ammonia only $\frac{1}{100}$ in. thick will decrease the rate of heat transfer to about 1 per cent of the amount that could be obtained if the surface were subject to pure conduction only. The same thing is true of liquids, for if liquids are present a surface film is formed which offers a great resistance to the flow of heat. If the design is such that one half of the cross section of the pipe is filled with liquid that surface not submerged is going to be more effective than the remainder. The bleeder type of condenser, for example, is illustrative of a partially flooded condenser condition as the liquid accumulates until the velocity of gas flow makes the liquid move upwards to a drip connection, or the head is developed sufficient to cause a flow downwards against the flow of the gas. The flooded condenser is designed for more or less filling of the pipes with liquid and the surface not submerged is made highly effective by the increased velocity of the gas flow occasioned by the resulting decrease in the cross-sectional area of the pipe. This velocity is paid for by an increase in the pressure of discharge of the ammonia from the compressor. Tests by Ophuls and Greene and by Shipley[6] have indicated that the *best condenser* is that which drains the condensate most promptly and which has as direct a path from the condenser to the receiver for the liquid as it is practical to obtain. The tests referred to refute absolutely the value of the *flooded type* of condenser of the atmospheric, the double pipe or the shell and tube design and indicate that the value of the bleeder atmospheric condenser cannot be very great from the standpoint of getting counterflow. There may be an advantage, however, in increasing the effectiveness of the heat transfer surface by a prompter removal of the condensate over that which would be possible in the case of the common type condenser and to a lesser extent in the double-pipe counterflow condenser. In all pipe condensers, however, there appears to be an objection due to the

[6] Shipley, National Association of Practical Refrigerating Engineers, 1923.

slowness of the removal of the liquid, and in the resulting non-effectiveness of a portion of the surface.

*The Common Atmospheric Condenser.*—The common atmospheric condenser is shown in Fig. 90. It usually has at least three pipes at the bottom for the removal of the superheat, and then the gas passes to the top of the condenser. As the top pipe has the coldest water, practically the entire load of condensation is taken by the upper four pipes, except when the condenser has a heavy load due to a higher suction pressure or when relatively large quantities of warm water are showered over the condenser. The liquid produced passes from one pipe to another and finally drains off at the low point. There is small likelihood of liquid sub-cooling because the temperature of the liquid leaving the last pipes subject to saturation conditions are in metal contact with relatively warm water. The pipe surface in the lower condensation pipes is partly non-effective, due to the presence of liquid ammonia which has to pass through all the pipes to the bottom in order to get out, but this is not such an important factor as the lower pipe surface is not very much needed for liquefaction purposes. In all likelihood six pipes in the region of saturation are plenty and the 16, 18, 24 and 30 pipe condensers are uneconomical of pipe surface as far as *condensing* ammonia is concerned. The surface usually supplied is from 8 to 10 sq. ft. per ton of refrigeration.

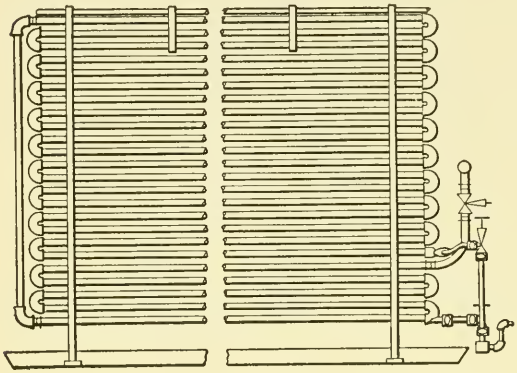


Fig. 90.—Common Atmospheric Ammonia Condenser.

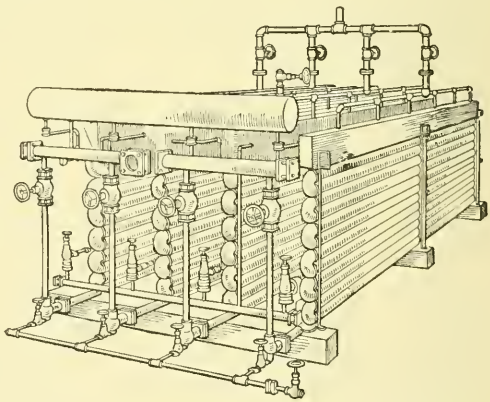


Fig. 91.—Four Stands of Atmospheric Condensers.

*The Bleeder Condenser.*—The bleeder type atmospheric condenser is

shown in Fig. 92. The hot gas always enters at the bottom and ascends so that the upper pipe is, to all intents and purposes, a dead end. The coldest water is showered over the entire length of the upper pipe and

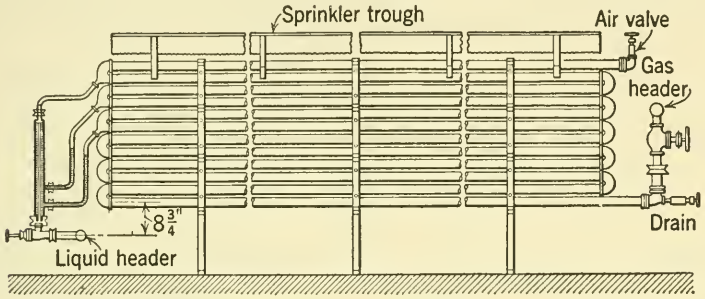



Fig. 92.—The Bleeder Type Condenser.

this is where the greatest load on the condenser occurs. Two and sometimes three tap offs are provided for the removal of the liquid. Tests indicate little difference in the static pressure between the gas entering the bottom and the gas in the top pipe, and this slight difference is

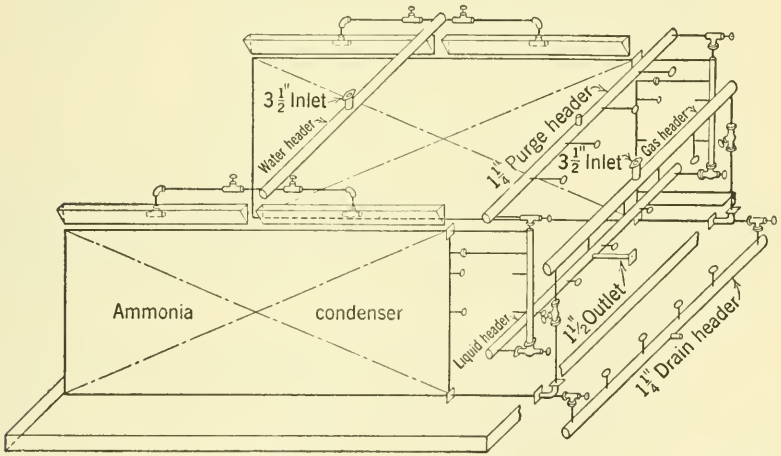



Fig. 93.—Six Stands, Atmospheric Ammonia Condensers, with Headers and Water Distributing Troughs.

easily accounted for in the resistance to the flow of the gas through the pipes and fittings. From 6 to 10 sq. ft. of surface are provided per ton of refrigeration as a rule.

*The Flooded Atmospheric Condenser.*—The flooded atmospheric condenser, (Fig. 94) uses an ejector nozzle, as shown in the insert, in order to

pick up the liquid condensate required for its proper performance. The lower pipes are practically similar in effect to those shown in the common atmospheric condenser. It would appear that, after the liquid is picked up by the gas, and is carried in more or less large plugs up to the upper pipe, on arrival at the upper part of the condenser the liquid separates from the gas and the two flow independent of one another except for the contact at the top of the liquid surface. The velocity of the gas in 2-in. pipe is small, being at usual (and standard) operating conditions about 8 ft. per second average velocity for the cross section of the pipe. With the flooded condenser the net area available for the passage of the gas would be one-half or possibly one-quarter of this amount and the velocity of the gas would be increased in proportion to the decrease of cross-sectional area. Remarkable performances have been obtained

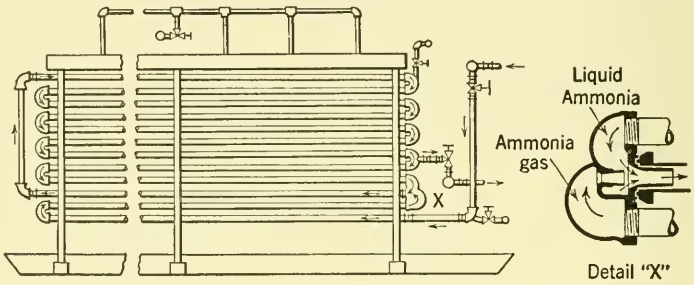



Fig. 94.—The Flooded Type Ammonia Condenser.

at times with this condenser, but when placed in parallel with other flooded condensers it is next to impossible to keep all the condensers performing equally. This condenser, also, is not one that will give the minimum head pressure, and this pressure may increase 20 lb. without any apparent cause.

*The Double-pipe Condenser.*—The atmospheric condenser needs to be used on the roof or otherwise located so as to expose freely the wetted surfaces to the air, which should pass through the stands freely. Where convenience or necessity requires the condenser to be located in the compressor room some other form or design must be used. For such conditions as in theatres, office buildings, hotels, etc., the double-pipe condenser needs to be used and (up to within the last few years) also for small installations.

The double-pipe condenser (Fig. 95), takes the gas at the top and the water enters at the bottom. The action of this condenser is counter-flow, the refrigerant being inside the outer and the water inside the inner pipe. The water is therefore restrained, passing definitely from

pipe to pipe and has the number of passes corresponding to the number of pipes high of the stand. Having a fixed path the water velocity can be varied at will, something which is not true of any of the atmospheric condensers or of the vertical shell and tube condenser. The

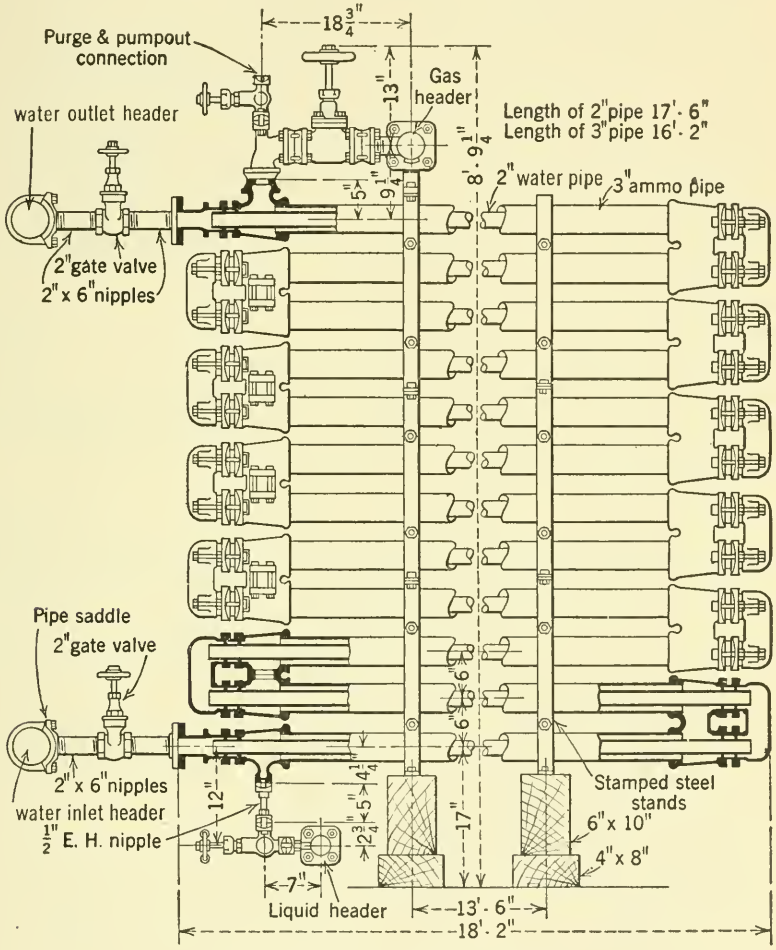



Fig. 95.—The Double-pipe Condenser.

capacity of the condenser can then be varied at will by changing the amount of water pumped through the condenser, although—incidentally—the friction head to be overcome by the water circulating pump increases very decidedly with an increase of the water velocity. The small cross-sectional area for the passage of the gas on entering the condenser causes some shock and vibration of the condenser and this

needs to be guarded against. There are also a large number of special fittings and special gaskets so that the condenser may have trouble from gas leaks unless the pipe and fittings are welded throughout, a resort which is becoming common in the case of the carbonic condenser.

As the gas enters the top and passes downward through the stand liquefaction takes place, but by far the greatest amount of the condensate takes place in the lower two pipes incidental with the coldest water. However, the sub-cooling effect due to the counterflow does not appear to be borne out in practice unless resistance to free draining or some other cause makes the liquid accumulate in the lower pipes. Sub-cooling is therefore not an inherent advantage of the double-pipe condenser, but indicates faulty piping or something not as it should be in a correctly designed and erected apparatus. This condenser is frequently a favorite for it has a large value of heat transfer, and is excellent in its performance, provided the water is reasonably free from scale-forming impurities. Unlike the atmospheric and the shell and tube condenser there is no appreciable splash of the water off of the condenser surfaces, which is a factor of extreme importance at times. The double-pipe condenser usually has from 6 to 10 sq. ft. per ton of refrigeration.

*The Shell and Tube Condenser.*—The shell and tube condenser (Fig. 96) is the latest and, from the design point of view, the best ammonia condenser built at the present time. Except for the small self-contained units, where the condenser is multi-pass and is placed horizontally, the shell and tube condenser is vertical and is designed for 2-in. tubes. The length of the tubes may vary from 6 ft.-0 in. in the 5- or 6-ton size to 10, 12 or 16 ft., depending in the larger sizes on the head room in the plant and the desires of the purchaser. The tubes are rolled into the tube sheet in a manner similar to a boiler tube. The tubes are laid out on approximately $2\frac{5}{8}$-in. centers, with the rows of tubes at 60 degrees with one another similar to steam surface condenser practice. Open lanes can be provided and the gas enters in the center of the shell as a rule. As the condenser applied to refrigeration is comparatively new it is not clear what will be the outcome, but there is every indication that it will be the most popular design, especially in the larger sizes. It is compact to a remarkable degree as a 100-ton condenser needs to be only about 46 in. in diameter and 14 ft. to 2 in. total height and it is free from the multiplicity of valves and fittings, although there is a chance for the tubes to require additional rolling to maintain tightness.

The water distribution is obtained by means of saw-tooth distributors, by a distribution designed to give the condensing water a whirling action or other, the objective being to keep the water on the inside surface of the tubes and yet limit the amount of the thickness of the water to a

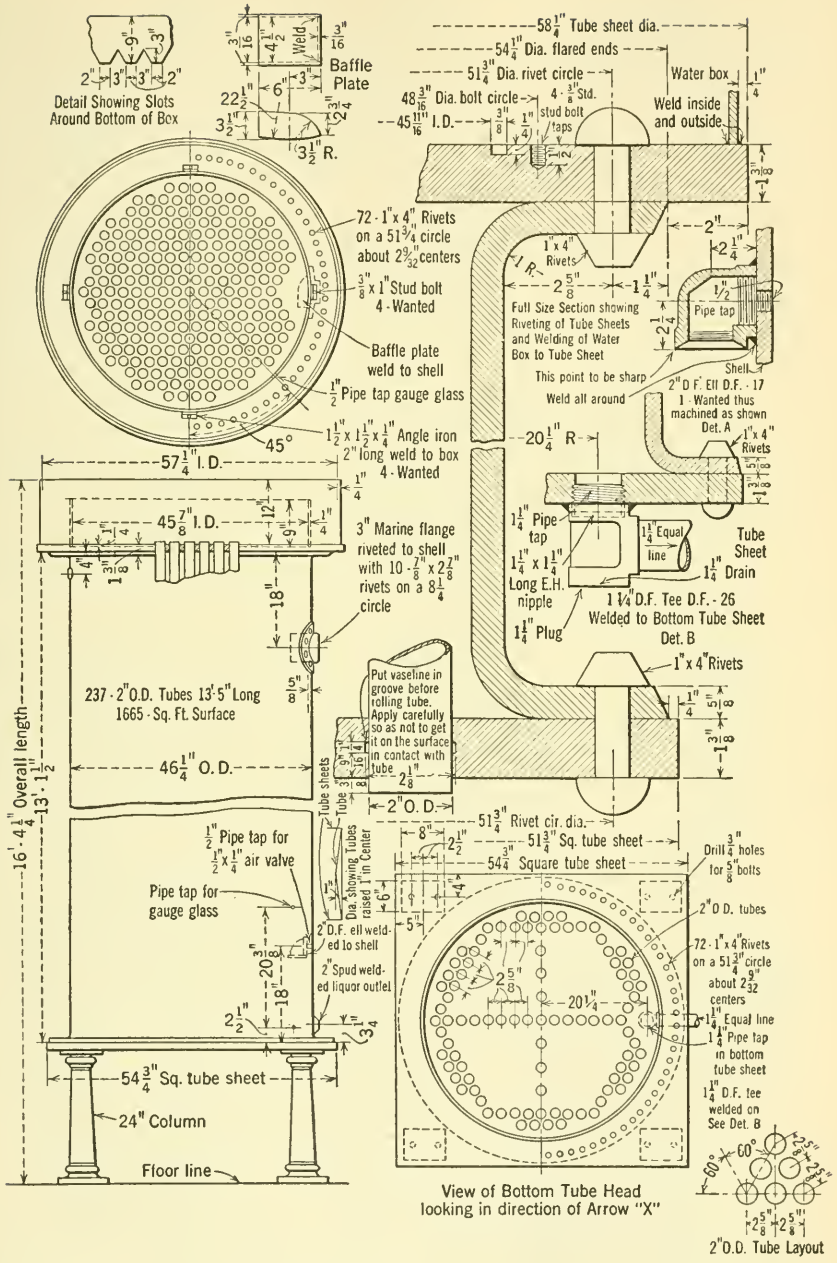

Detail Showing Slots Around Bottom of Box
Baffle Plate
Weld
3½" R.
58¼" Tube sheet dia.
54¼" Dia. flared ends
51¾" Dia. rivet circle
48⅝" Dia. bolt circle
45 11/16" I.D.
4 - ⅜" Std. stud bolt taps
Water box
Weld inside and outside
72 - 1" x 4" Rivets on a 51¾" circle about 2 9/32" centers
⅜" x 1" Stud bolt 4 - Wanted
Baffle plate weld to shell
½" Pipe tap gauge glass
45°
1½" x 1½" x ¼" Angle iron 2" long weld to box 4 - Wanted
57¼" I.D.
45⅞" I.D.
3" Marine flange riveted to shell with 10 - ⅞" x 2⅞" rivets on a 8¼" circle
18"
237 - 2" O.D. Tubes 13'-5" Long
1665 - Sq. Ft. Surface
16'-4¼" Overall length
13'-1½"
46¼" O.D.
Put vaseline in groove before rolling tube. Apply carefully so as not to get it on the surface in contact with tube
2" O.D.
Tube sheets
½" Pipe tap for ½" x ¼" air valve
Pipe tap for gauge glass
2" D.F. ell welded to shell
2" Spud welded liquor outlet
20⅜"
18"
2½"
3¼"
54¾" Sq. tube sheet
24" Column
Floor line
1" x 4" Rivets
Full Size Section showing Riveting of Tube Sheets and Welding of Water Box to Tube Sheet
Pipe tap
Shell
This point to be sharp
Weld all around
2" D.F. Ell D.F. - 17
1 - Wanted thus machined as shown Det. A
20¼" R
1¼" Pipe tap
1¼" x 1¼" Long E.H. nipple
1¼" Plug
1¼" Equal line
Tube Sheet
1¼" Drain
1 ¼" D.F. Tee D.F. - 26
Welded to Bottom Tube Sheet Det. B
Dia. Showing Tubes raised 1 in Center
51¾" Rivet cir. dia.
51¾" Sq. tube sheet
54¾" Square tube sheet
Drill ¾" holes for ⅝" bolts
2" O.D. tubes
72 - 1" x 4" Rivets on a 51¾" circle about 2 9/32" centers
1¼" Equal line
1¼" Pipe tap in bottom tube sheet
1¼" D.F. tee welded on See Det. B
View of Bottom Tube Head looking in direction of Arrow "X"
60°
2" O.D. Tube Layout

FIG. 96.—The Shell and Tube Condenser.

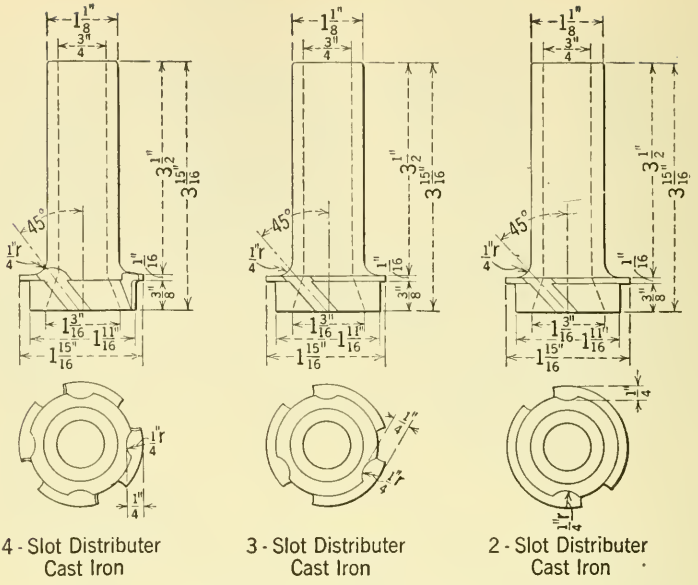



FIG. 97.—Water Distributors for Shell and Tube Condensers.

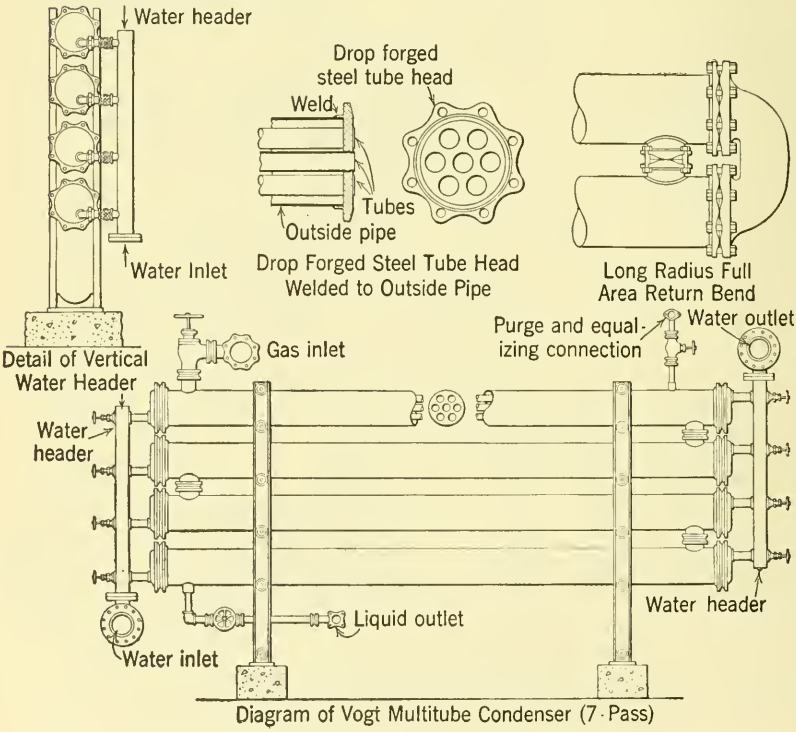



FIG. 98.—The Multitube Ammonia Condenser.

thin film. As the water enters the tubes at the top the greater part of the liquefaction takes place in the upper part of the condenser and the condensate flows by gravity down the tubes. There are, therefore, two liquid films—of liquid ammonia on the outside and of water on the inside of the tubes. However there is little tendency for the formation of an inert film on the surfaces and in consequence the rate of heat transfer is large and there is a prompt freeing of the surface of the liquid condensate and the heat transfer surface remains effective. The gas from the compressor may enter lanes arranged for it in the condenser so that the gas shock is negligible. Arranged in batteries there is no difficulty in making each condenser take its proper share of the load. A safety valve on the condenser will safeguard against excess pressure, which otherwise might cause rupture of the shell. The shell and tube condenser is designed at present (1927) for from 12 to 15 sq. ft. of heat transfer surface per ton of refrigeration, but tests by the author indicate that this amount is excessive.

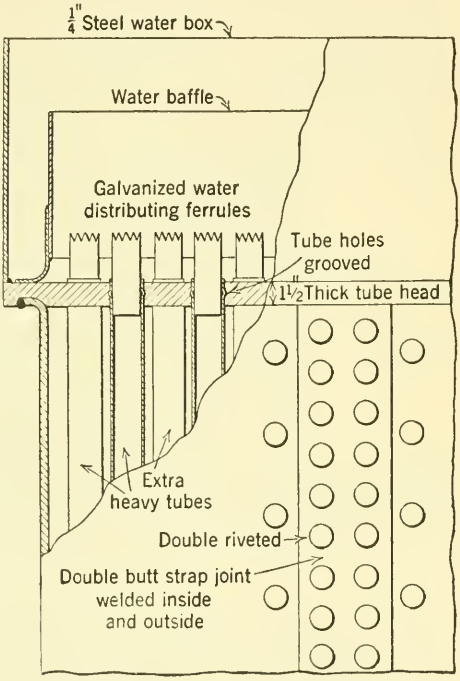



FIG. 99.—The Shell and Tube Condenser.

**Heat Removed by Condenser.**—The heat removed by the condenser is equal to the refrigerating effect plus the heat equivalent of the work done on the gas during the compression. This may be expressed *per ton of refrigeration per minute* as follows:

$Q_c$ = the indicated horse power times 42.44 plus 200

$$= \frac{4.713}{E_r}\left[\frac{i_4 - i_3}{i_3 - i_2}\right] \times 42.44 + 200 = 200\left[\left(\frac{i_4 - i_3}{i_3 - i_2} \times \frac{1}{E_r}\right) + 1\right]$$

Where $i_4 - i_3$ is equal to the increase of the total heat per pound of ammonia during the compression, $i_3 - i_2$ is the refrigerating effect per pound of ammonia and $E_r$ is the real volumetric efficiency of the compressor. With an expansion valve $i_1 = i_2$ (Fig. 1).

**Water Requirements of The Condenser.**—Amount of condenser water in gallons per ton per minute

$$= \frac{Q_c}{8.33 \times (t_2 - t_1)} = \frac{200\left(\frac{i_4 - i_3}{i_3 - i_1}\right) \times \left(\frac{1}{E_r} + 1\right)}{8.33 \times (t_2 - t_1)}$$

where the term $(t_2 - t_1)$ represents the rise in temperature of the condensing water.

Heat removed theoretically from different parts of the condenser per ton per minute.

*a.* $$Q_{\text{latent}} = \frac{200 \times r}{(i_3 - i_7)}$$

where

$$r = i_6 - i_1 \text{ (Fig 1)}$$

The points *1* and *7* will coincide, unless a separate aftercooler is used, if no inert gases are present in the condenser.

*b.* $$Q_{\text{aftercooler}} = \frac{200 \times (i_1 - i_7)}{(i_3 - i_7)}$$

*c.* $$Q_{\text{superheat}} = \frac{200 \times (i_4 - i_6)}{(i_3 - i_7)}$$

and the *average mean temperature* difference is:

$$t_m = \frac{Q_l\, t_l + Q_a\, t_a + Q_s\, t_s}{Q_c}$$

where $t_a$, $t_b$, and $t_c$ are the average temperature differences in the latent heat, the aftercooler and the superheat regions respectively.

**The Effect of Inert Gases in the Condenser.**—In steam condenser practice using the surface condenser it has been found that the presence of *air* in the steam has the effect of reducing very greatly the coefficient of heat transfer and that this value could be from 5 to 10 times as great as is found in actual practice if air could be eliminated. The effect of this air in the surface condenser is to mix with the condensible gases, and as condensation continues, to become more and more concentrated as it approaches the surface of the tubes. The result of this is the development of a non-condensible gas film on the metal surface. As the amount of air in the condenser increases the thickness of this air film increases, resulting in a lowering of the value of the coefficient of heat transfer. In addition to the loss of surface efficiency the total pressure of the condenser is increased in accordance with Dalton's law which says that the total pressure is the sum of the partial pressures,

and that the partial pressures exerted by each gas or vapor are those which would be caused by each occupying the volume by themselves. In other words,

$$p = p_1 - p_2$$

where $p_1$ = partial pressure of the air

$p_2$ = partial pressure of (in this case) the ammonia

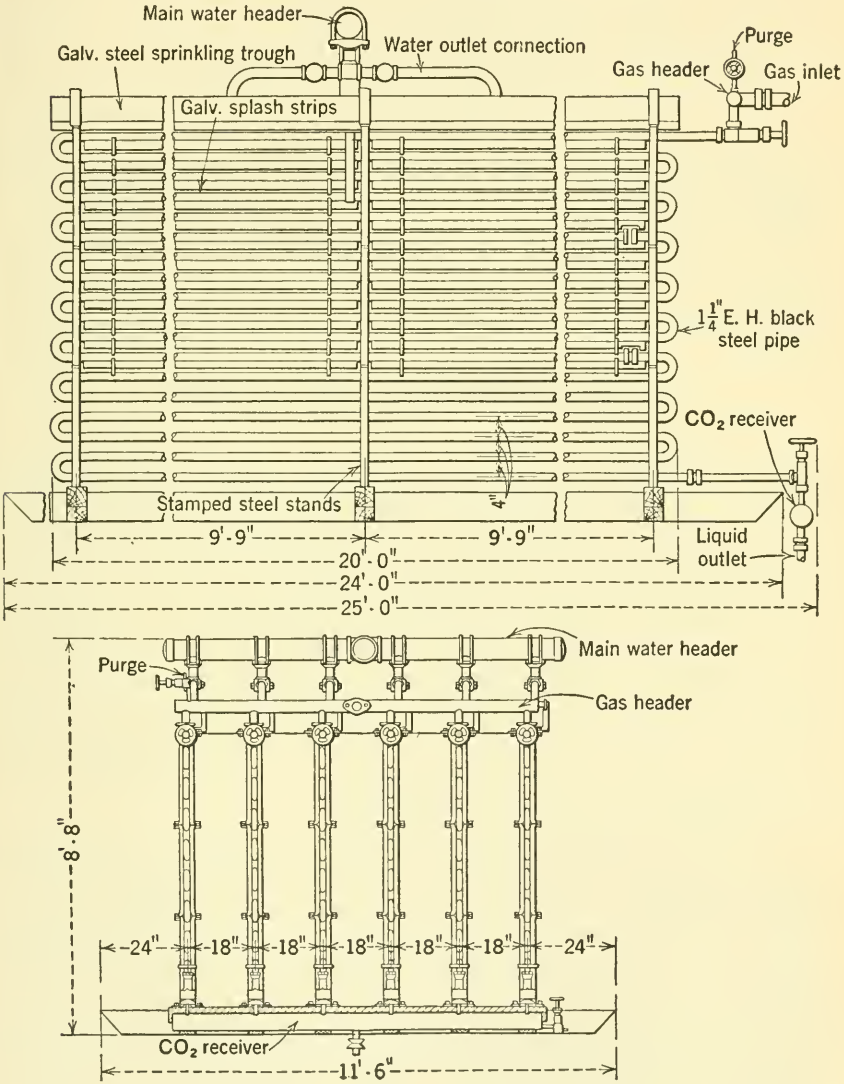



FIG. 100.—Atmospheric Type $CO_2$ Condenser.

In the case of the ammonia condenser the partial pressure of the air in pounds per square inch could be calculated, if the volume of the condenser were known and the temperature and the weight of the air present could be determined. Then by means of the characteristic equation for a perfect gas (air),

$$pv = MBT$$

where $M$ = weight of gas in pounds;
$B$ = gas constant = 53.3 for air;
$T$ = absolute temperature of the air in degrees F.;
$v$ = volume in cubic feet of the gas under a pressure $p$ lb. per square inch absolute.

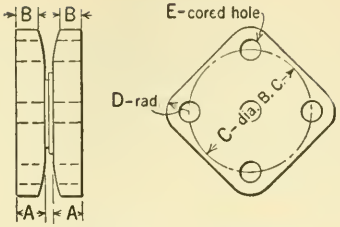
The partial pressure of the ammonia is that corresponding to the temperature during liquefaction, as given in the ammonia tables.

For example a stand of 12–2 in. pipes, 20 ft. long, has a volume of approximately $\frac{12 \times 20}{42.9}$ cu. ft. and the weight of air that would exert a pressure of 10 lb. per square inch at a temperature of 70 deg. F. is

$$M = \frac{pv}{BT} = \frac{10 \times 12 \times 20 \times 144}{42.9 \times 53.3 \times 530} = 0.285 \text{ lb.}$$

As the refrigerating system is a closed one and the pressure is normally always greater than atmospheric pressure there is not much reason for air to get into the condenser. However there is always some air remaining in the system on charging, or whenever repairs are made to any part of the plant and undoubtedly some air enters the compressor when it is operated at a suction pressure below that of the atmosphere. The action of the boiling of the refrigerant in the evaporating coils, and the velocity of the gas flow into the compressor and the condenser results in the picking up of the air and the final depositing of it in the condenser. Any non-condensible gas which enters the condenser will stay there until it is pumped out or leaks out (with the ammonia) and as it accumulates the pressure will increase.

The old method of purging non-condensible gases from the condenser was to open the purge valve and permit some of the gas to escape to the atmosphere. However to save ammonia some special device is necessary, for if air once gets into the condenser it is next to impossible to get rid of it, by purging, without loss of ammonia. The best method of purging is to permit a slight flow of gas from the condenser, or from the top of the liquid receiver, either continually or intermittently and to refrigerate this gas by means of a special evaporating coil. The gas permitted to escape under these conditions will be air in most part,



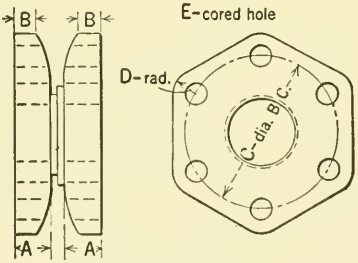

$CO^2$ Flange - 4-Bolt Type

*a*

$CO_2$ Flange
4-Bolt Type

| Size | Symbol Number | A | B | C | D | E | Size of Bolts |
|---|---|---|---|---|---|---|---|
| $\frac{3}{4}$ | 1024-B | $\frac{7}{8}$ | $\frac{5}{8}$ | $3\frac{1}{8}$ | $\frac{5}{8}$ | $\frac{5}{8}$ | $\frac{1}{2}\times 2\frac{7}{8}$ |
| 1 | 1024-C | $1\frac{1}{16}$ | $\frac{3}{4}$ | $3\frac{5}{8}$ | $\frac{3}{4}$ | $\frac{3}{4}$ | $\frac{5}{8}\times 3\frac{3}{8}$ |
| $1\frac{1}{4}$ | 1024-D | $1\frac{3}{8}$ | $\frac{3}{4}$ | $4\frac{1}{8}$ | $\frac{3}{4}$ | $\frac{3}{4}$ | $\frac{5}{8}\times 3\frac{1}{8}$ |
| $1\frac{1}{2}$ | 1024-E | $1\frac{1}{8}$ | $\frac{3}{4}$ | $4\frac{1}{2}$ | $\frac{3}{4}$ | $\frac{3}{4}$ | $\frac{5}{8}\times 3\frac{1}{2}$ |
| 2 | 1024-F | $1\frac{1}{4}$ | $\frac{15}{16}$ | $5\frac{1}{8}$ | $\frac{7}{8}$ | $\frac{7}{8}$ | $\frac{3}{4}\times 3\frac{7}{8}$ |

All dimensions given in inches unless otherwise specified.




$CO^2$ Flange - 6-Bolt Type

*b*

$CO_2$ Flange
6-Bolt Type

| Size | Symbol Number | A | B | C | D | E | Size of Bolts |
|---|---|---|---|---|---|---|---|
| $2\frac{1}{2}$ | 1292-F | $1\frac{5}{8}$ | 1 | $6\frac{1}{2}$ | 1 | $\frac{7}{8}$ | $\frac{3}{4}\times 5\frac{1}{4}$ |
| 3 | 1293-F | $1\frac{3}{4}$ | $1\frac{1}{8}$ | $7\frac{3}{4}$ | 1 | 1 | $\frac{7}{8}\times 5$ |
| 4 | 1294-F | $2\frac{1}{4}$ | $1\frac{1}{4}$ | $9\frac{1}{2}$ | $1\frac{1}{8}$ | $1\frac{1}{4}$ | $1\frac{1}{8}\times 6$ |

All dimensions given in inches unless otherwise specified.

Fig. 101.—Carbonic Fittings—Flanges.

although the volume of gas in the holder contains as much ammonia as would be present at that temperature were there no air present, and the weight of ammonia purged would be the total volume of gas purged (at the pressure of the tank) divided by specific volume of ammonia at the temperature maintained by the cooling coils. If one cubic foot is purged and the temperature is 0 deg. F. then $\frac{1}{9.12} = 0.11$ lb. of ammonia is lost, but if the temperature is 32 deg. F. then $\frac{1}{4.64} = 0.216$ lb. will be the amount of ammonia lost.

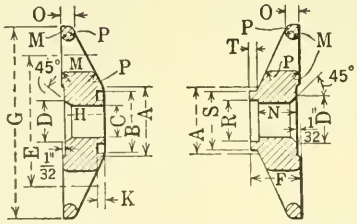
TABLE 19

RECEIVERS—HIGH-PRESSURE OIL SEPARATORS—AMMONIA ACCUMULATORS

| Tons Ice Making | Diameter, Inches | Length | | Pipe Connections, Inches | Weight, Pounds |
|---|---|---|---|---|---|
| | | Feet | Inches | | |
| **Ammonia Receivers** | | | | | |
| 1 and 2 | 12 | 4 | 0 | $\frac{1}{2}$ | 290 |
| 3 | 12 | 6 | 0 | $\frac{1}{2}$ | 375 |
| 4 | 12 | 8 | 0 | $\frac{1}{2}$ | 460 |
| 5 and 6 | 16 | 7 | 0 | $\frac{1}{2}$ | 475 |
| 8 to 20 | 20 | 7 | 0 | $\frac{3}{4}$ | 650 |
| 25 | 20 | 10 | 0 | 1 | 850 |
| 30 to 50 | 20 | 16 | 0 | $1\frac{1}{4}$ | 1200 |
| 60 to 100 | 24 | 14 | 0 | 2 | 1600 |
| 125 to 200 | 24 | 16 | 0 | 2 | 1770 |
| **H. P. Oil Separators** | | | | | |
| 1 to 12 | 12 | 3 | 0 | $1\frac{1}{4}$ and 2 | 220 |
| 15 to 35 | 16 | 4 | 0 | $2\frac{1}{2}$ and 3 | 370 |
| 40 to 60 | 20 | 5 | 0 | $3\frac{1}{2}$ and 4 | 600 |
| 75 to 100 | 24 | 5 | 0 | 5 and 6 | 890 |
| **Ammonia Accumulators** | | | | | |
| 1 to 12 | 12 | 3 | 0 | $1\frac{1}{4}$ and 2 | 285 |
| 15 to 25 | 16 | 4 | 0 | $2\frac{1}{2}$ and 3 | 400 |
| 30 to 75 | 20 | 5 | 0 | $3\frac{1}{2}$ and 4 | 650 |
| 100 to 200 | 24 | 5 | 0 | 5 and 6 | 910 |

# TABLE 20

## Dimensions of Ammonia Companion Flanges. Tongue and Groove—Straight Backs (Crane Co.)

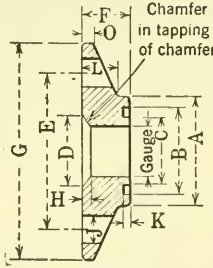



Section thru groove flange

$\frac{1}{4}$" to 2" Forged Steel

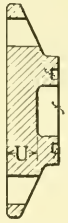



Section thru tongue flange

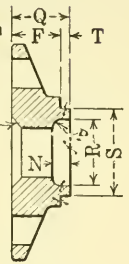



Groove

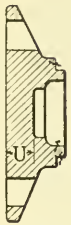
$2\frac{1}{2}$" Up Mall. Iron




Blind Groove




Tongue




Blind Groove

Blind flanges are made by filling in patt. as shown by letter "U"

6" flange has 12 holes

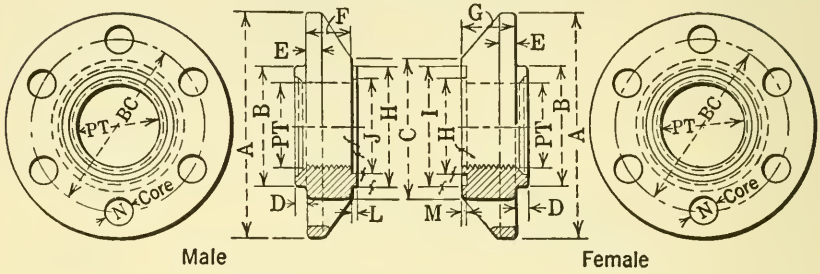
All Dimensions Given in Inches

| | A | B | C | D | E | F | G | H | J | K | L | M | N | O | P | Q | R | S | T | U | V |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| $\frac{1}{4}$ | $1\frac{1}{8}$ | 1 | $\frac{9}{16}$ | $\frac{21}{32}$ | $2\frac{3}{8}$ | $\frac{7}{8}$ | $3\frac{5}{8}$ | $\frac{21}{32}$ | $\frac{5}{8}$ | $\frac{1}{8}$ | ..... | $\frac{1}{8}$ | $\frac{21}{32}$ | $\frac{9}{32}$ | $\frac{5}{32}$ | ..... | $\frac{19}{32}$ | $\frac{31}{32}$ | $\frac{3}{16}$ | ..... | $1\frac{5}{8}$ |
| $\frac{3}{8}$ | $1\frac{1}{4}$ | $1\frac{1}{8}$ | $\frac{11}{16}$ | $\frac{13}{32}$ | $2\frac{9}{16}$ | $\frac{31}{32}$ | $3\frac{13}{16}$ | $\frac{3}{4}$ | $\frac{5}{8}$ | $\frac{1}{8}$ | ..... | $\frac{1}{8}$ | $\frac{3}{4}$ | $\frac{9}{32}$ | $\frac{5}{32}$ | ..... | $\frac{23}{32}$ | $1\frac{3}{32}$ | $\frac{3}{16}$ | ..... | $1\frac{3}{4}$ |
| $\frac{1}{2}$ | $1\frac{3}{8}$ | $1\frac{1}{4}$ | $\frac{27}{32}$ | $\frac{31}{32}$ | $2\frac{3}{4}$ | $1\frac{1}{16}$ | 4 | $\frac{27}{32}$ | $\frac{5}{8}$ | $\frac{1}{8}$ | ..... | $\frac{1}{8}$ | $\frac{27}{32}$ | $\frac{9}{32}$ | $\frac{5}{32}$ | ..... | $\frac{7}{8}$ | $1\frac{7}{32}$ | $\frac{3}{16}$ | ..... | 2 |
| $\frac{3}{4}$ | $1\frac{5}{8}$ | $1\frac{1}{2}$ | $1\frac{3}{32}$ | $1\frac{7}{32}$ | $3\frac{1}{4}$ | $1\frac{3}{32}$ | $4\frac{3}{4}$ | $\frac{29}{32}$ | $\frac{3}{4}$ | $\frac{1}{8}$ | ..... | $\frac{1}{8}$ | $\frac{29}{32}$ | $\frac{9}{32}$ | $\frac{5}{32}$ | ..... | $1\frac{1}{8}$ | $1\frac{15}{32}$ | $\frac{3}{16}$ | ..... | $2\frac{1}{4}$ |
| 1 | 2 | $1\frac{13}{16}$ | $1\frac{11}{32}$ | $1\frac{15}{32}$ | $3\frac{1}{4}$ | $1\frac{7}{32}$ | $3\frac{9}{16}$ | 1 | $\frac{5}{8}$ | $\frac{1}{8}$ | ..... | $\frac{5}{32}$ | 1 | $\frac{9}{32}$ | ..... | ..... | $1\frac{1}{8}$ | $1\frac{25}{32}$ | $\frac{3}{16}$ | | |
| $1\frac{1}{4}$ | $2\frac{7}{16}$ | $2\frac{1}{4}$ | $1\frac{21}{32}$ | $1\frac{13}{16}$ | $3\frac{3}{4}$ | $1\frac{11}{32}$ | $3\frac{15}{16}$ | $1\frac{1}{16}$ | $\frac{3}{4}$ | $\frac{3}{16}$ | ..... | $\frac{5}{32}$ | $1\frac{1}{16}$ | $\frac{5}{16}$ | ..... | ..... | $1\frac{11}{16}$ | $2\frac{7}{32}$ | $\frac{1}{4}$ | ..... | ..... |
| $1\frac{1}{2}$ | $2\frac{11}{16}$ | $2\frac{1}{2}$ | $1\frac{29}{32}$ | $2\frac{3}{32}$ | $4\frac{1}{2}$ | $1\frac{13}{32}$ | $4\frac{11}{16}$ | $1\frac{1}{8}$ | $\frac{3}{4}$ | $\frac{3}{16}$ | ..... | $\frac{5}{32}$ | $1\frac{1}{8}$ | $\frac{11}{32}$ | ..... | ..... | $1\frac{15}{16}$ | $2\frac{15}{32}$ | $\frac{1}{4}$ | ..... | ..... |
| 2 | $3\frac{5}{16}$ | $3\frac{1}{8}$ | $2\frac{13}{32}$ | $2\frac{19}{32}$ | 5 | $1\frac{7}{16}$ | $5\frac{1}{16}$ | $1\frac{5}{32}$ | $\frac{7}{8}$ | $\frac{3}{16}$ | ..... | $\frac{3}{16}$ | $1\frac{5}{32}$ | $\frac{3}{8}$ | ..... | ..... | $2\frac{7}{16}$ | $3\frac{3}{32}$ | $\frac{1}{4}$ | | |
| $2\frac{1}{2}$ | $4\frac{1}{4}$ | $3\frac{11}{16}$ | $2\frac{29}{32}$ | $3\frac{5}{64}$ | $5\frac{7}{8}$ | $1\frac{7}{16}$ | $5\frac{15}{16}$ | $\frac{1}{4}$ | $\frac{7}{8}$ | $\frac{1}{4}$ | $1\frac{1}{4}$ | $\frac{3}{16}$ | $\frac{21}{32}$ | $\frac{3}{8}$ | $\frac{9}{32}$ | $1\frac{7}{8}$ | $2\frac{15}{16}$ | $3\frac{21}{32}$ | $\frac{7}{16}$ | 1 | |
| 3 | $4\frac{7}{8}$ | $4\frac{5}{16}$ | $3\frac{17}{32}$ | $3\frac{3}{4}$ | $6\frac{5}{8}$ | $1\frac{9}{16}$ | $8\frac{1}{4}$ | $\frac{17}{64}$ | $\frac{7}{8}$ | $\frac{1}{4}$ | $1\frac{3}{8}$ | $\frac{3}{16}$ | $\frac{45}{64}$ | $\frac{3}{8}$ | $\frac{9}{32}$ | 2 | $3\frac{9}{16}$ | $4\frac{9}{32}$ | $\frac{7}{16}$ | 1 | |
| $3\frac{1}{2}$ | $5\frac{1}{2}$ | $4\frac{7}{8}$ | $4\frac{1}{32}$ | $4\frac{1}{4}$ | $7\frac{1}{4}$ | $1\frac{5}{8}$ | 9 | $\frac{19}{64}$ | $\frac{7}{8}$ | $\frac{1}{4}$ | $1\frac{7}{16}$ | $\frac{5}{16}$ | $\frac{43}{64}$ | $\frac{3}{8}$ | $\frac{9}{32}$ | $2\frac{1}{16}$ | $4\frac{1}{16}$ | $4\frac{27}{32}$ | $\frac{7}{16}$ | $1\frac{1}{16}$ | |
| 4 | $6\frac{1}{8}$ | $5\frac{1}{2}$ | $4\frac{17}{32}$ | $4\frac{25}{32}$ | $7\frac{7}{8}$ | $1\frac{11}{16}$ | 10 | $\frac{5}{16}$ | $\frac{7}{8}$ | $\frac{1}{4}$ | $1\frac{1}{2}$ | $\frac{3}{16}$ | $\frac{21}{32}$ | $\frac{7}{16}$ | $\frac{5}{16}$ | $2\frac{1}{8}$ | $4\frac{9}{16}$ | $5\frac{15}{32}$ | $\frac{7}{16}$ | $1\frac{1}{8}$ | |
| 5 | $7\frac{5}{8}$ | $6\frac{7}{8}$ | $5\frac{19}{32}$ | $5\frac{7}{8}$ | $9\frac{1}{4}$ | $1\frac{27}{32}$ | 11 | $\frac{19}{64}$ | $\frac{7}{8}$ | $\frac{1}{4}$ | $1\frac{5}{8}$ | $\frac{3}{16}$ | $\frac{47}{64}$ | $\frac{1}{2}$ | $\frac{5}{16}$ | $2\frac{9}{32}$ | $5\frac{5}{8}$ | $6\frac{27}{32}$ | $\frac{7}{16}$ | $1\frac{3}{16}$ | |
| 6 | $8\frac{3}{4}$ | $7\frac{7}{8}$ | $6\frac{21}{32}$ | 7 | $10\frac{5}{8}$ | 2 | $12\frac{1}{2}$ | $\frac{3}{8}$ | $\frac{7}{8}$ | $\frac{1}{4}$ | $1\frac{11}{16}$ | $\frac{3}{16}$ | $\frac{3}{4}$ | $\frac{1}{2}$ | $\frac{5}{16}$ | $2\frac{7}{16}$ | $6\frac{11}{16}$ | $7\frac{27}{32}$ | $\frac{7}{16}$ | $1\frac{1}{4}$ | |

## TABLE 21

### Dimensions of Ammonia Companion Flanges

Tongue and Groove—Crane Co.

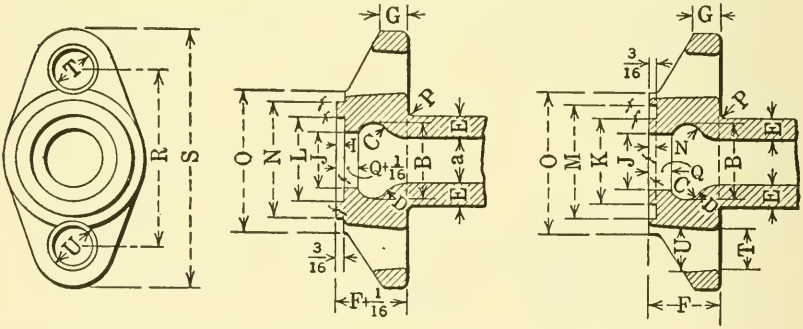



All Dimensions Given in Inches

| | A | B | C | D | E | F | G | H | J | K | L | M | N | O | P |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 7 | 14 | $9\frac{1}{2}$ | $8\frac{5}{8}$ | $9\frac{5}{8}$ | $9\frac{11}{16}$ | $8\frac{9}{16}$ | $2\frac{1}{16}$ | 1.61 | $1\frac{13}{16}$ | $1\frac{1}{2}$ | $7\frac{7}{8}$ | $\frac{1}{4}$ | $\frac{3}{16}$ | $11\frac{7}{8}$ | 12– $\frac{7}{8}$ |
| 8 | 15 | $10\frac{5}{8}$ | $9\frac{5}{8}$ | $10\frac{5}{8}$ | $10\frac{15}{16}$ | $9\frac{9}{16}$ | $2\frac{3}{16}$ | 1.71 | $1\frac{29}{32}$ | $1\frac{5}{8}$ | $8\frac{7}{8}$ | $\frac{1}{4}$ | $\frac{3}{16}$ | 13 | 12– $\frac{7}{8}$ |
| 10 | $17\frac{1}{2}$ | $12\frac{5}{8}$ | $11\frac{5}{8}$ | $13\frac{1}{8}$ | $13\frac{3}{16}$ | $11\frac{9}{16}$ | $2\frac{3}{8}$ | 1.92 | $2\frac{7}{64}$ | $1\frac{7}{8}$ | 11 | $\frac{1}{4}$ | $\frac{3}{16}$ | $15\frac{1}{4}$ | 16–1 |
| 12 | $20\frac{1}{2}$ | $14\frac{3}{4}$ | $13\frac{5}{8}$ | $15\frac{1}{8}$ | $15\frac{3}{16}$ | $13\frac{9}{16}$ | $2\frac{9}{16}$ | 2.12 | $2\frac{5}{16}$ | 2 | 13 | $\frac{1}{4}$ | $\frac{3}{16}$ | $17\frac{3}{4}$ | 16–$1\frac{1}{8}$ |

## TABLE 22

### Flange Dimensions of "York" Oval Flange Fittings

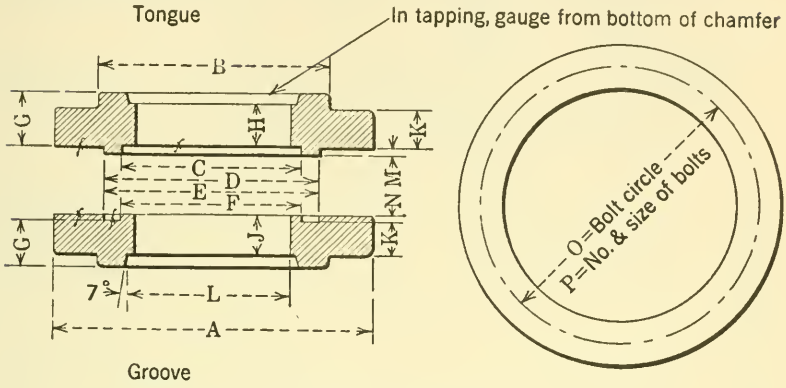



All Dimensions Given in Inches

| Size | *A* | *B* | *C* | *D* | *E* | *F* | *G* | *H* | *I* | *J* | *K* | *L* | *M* | *N* | *O* | *P* | *Q* | *R* | *S* | *T* | *U* |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| $\frac{1}{4}$ | $\frac{1}{2}$ | $\frac{3}{4}$ | $\frac{3}{16}$ | $\frac{1}{2}$ | $\frac{1}{4}$ | $\frac{7}{8}$ | $\frac{5}{16}$ | $\frac{1}{8}$ | $\frac{5}{32}$ | $1\frac{15}{32}$ | $\frac{7}{8}$ | $2\frac{9}{32}$ | $1\frac{3}{8}$ | $1\frac{11}{32}$ | $1\frac{3}{4}$ | $\frac{1}{16}$ | $\frac{3}{8}$ | $2\frac{1}{4}$ | $3\frac{1}{2}$ | $\frac{9}{16}$ | $\frac{5}{8}$ |
| $\frac{3}{8}$ and $\frac{1}{2}$ | $\frac{5}{8}$ | $1\frac{15}{16}$ | $\frac{1}{4}$ | $\frac{1}{2}$ | $\frac{5}{16}$ | 1 | $\frac{5}{16}$ | $\frac{1}{8}$ | $\frac{5}{32}$ | $1\frac{11}{16}$ | $1\frac{1}{4}$ | $1\frac{9}{32}$ | $1\frac{3}{4}$ | $1\frac{23}{32}$ | $2\frac{1}{8}$ | $\frac{1}{8}$ | $\frac{3}{8}$ | $2\frac{5}{8}$ | $3\frac{3}{4}$ | $\frac{9}{16}$ | $\frac{5}{8}$ |
| $\frac{3}{4}$ | $\frac{3}{4}$ | $1\frac{3}{16}$ | $\frac{5}{16}$ | $\frac{1}{2}$ | $\frac{5}{16}$ | $1\frac{1}{8}$ | $\frac{3}{8}$ | $\frac{1}{8}$ | $\frac{5}{32}$ | $1\frac{15}{16}$ | $1\frac{5}{16}$ | $1\frac{11}{32}$ | $1\frac{13}{16}$ | $1\frac{25}{32}$ | $2\frac{3}{16}$ | $\frac{1}{16}$ | $\frac{1}{2}$ | $2\frac{3}{4}$ | $4\frac{1}{8}$ | $\frac{11}{16}$ | $\frac{3}{4}$ |
| 1 | 1 | $1\frac{7}{16}$ | $\frac{3}{8}$ | $\frac{1}{2}$ | $\frac{5}{16}$ | $1\frac{1}{4}$ | $\frac{3}{8}$ | $\frac{1}{8}$ | $\frac{5}{32}$ | $1\frac{3}{16}$ | $1\frac{5}{8}$ | $1\frac{21}{32}$ | $2\frac{1}{8}$ | $2\frac{3}{32}$ | $2\frac{1}{2}$ | $\frac{1}{8}$ | $\frac{1}{2}$ | $3\frac{1}{8}$ | $4\frac{1}{2}$ | $\frac{11}{16}$ | $\frac{3}{4}$ |
| $1\frac{1}{4}$ | $1\frac{1}{4}$ | $1\frac{3}{4}$ | $\frac{7}{16}$ | $\frac{1}{2}$ | $\frac{5}{16}$ | $1\frac{3}{8}$ | $\frac{7}{16}$ | $\frac{1}{8}$ | $\frac{5}{32}$ | $1\frac{15}{32}$ | $1\frac{13}{16}$ | $1\frac{27}{32}$ | $2\frac{5}{16}$ | $2\frac{9}{32}$ | $2\frac{5}{8}$ | $\frac{1}{16}$ | $\frac{1}{2}$ | $3\frac{5}{16}$ | $4\frac{7}{8}$ | $\frac{11}{16}$ | $\frac{3}{4}$ |

## TABLE 23

### Dimensions of "York" Tongue and Groove Round Flanges

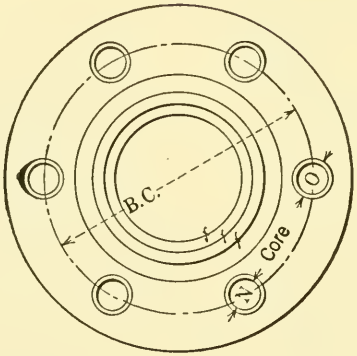


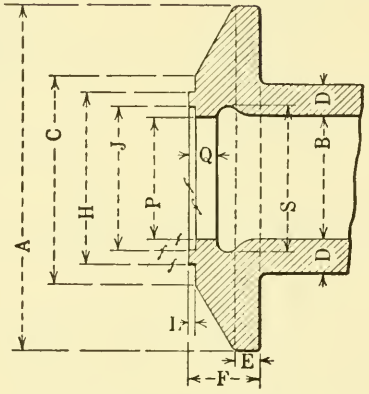

All Dimensions Given in Inches

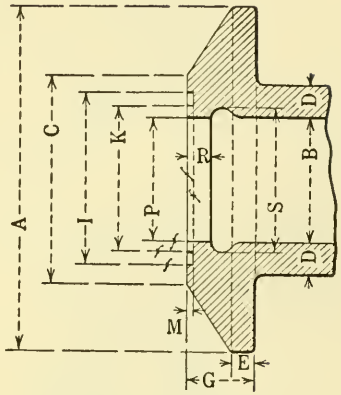
| Size, P.T. | *A* | *B* | *C* | *D* | *E* | *F* | *G* | *H* | *I* | *J* | *K* | *L* | *M* | *N*, Core | Bolts | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | | | | | | | | *B.C.* | No. | Size |
| 2½ | 7⅛ | 3⅝ | 4¼ | 3/8 | ½ | 1 7/16 | 1⅝ | 3¾ | 3 25/32 | 3 | 2 31/32 | 5/32 | ⅛ | 7/8 | 5½ | 6 | ¾×4 |
| 3 | 7⅞ | 4 5/16 | 4¾ | 3/8 | 9/16 | 1 9/16 | 1⅞ | 4⅜ | 4 13/32 | 3⅝ | 3 19/32 | 5/32 | ⅛ | 7/8 | 6⅛ | 6 | ¾×4¼ |
| 3½ | 8⅝ | 4⅞ | 5⅜ | 3/8 | 9/16 | 1⅝ | 1 13/16 | 4⅞ | 4 29/32 | 4⅛ | 4 3/32 | 5/32 | ⅛ | 7/8 | 6⅞ | 8 | ¾×4½ |
| 4 | 9⅜ | 5⅜ | 6⅛ | 3/8 | ⅝ | 1 11/16 | 1⅞ | 5½ | 5 17/32 | 4¾ | 4 23/32 | 5/32 | ⅛ | 7/8 | 7⅝ | 8 | ¾×4½ |
| 4½ | 9¾ | 5 13/16 | 6½ | 3/8 | ⅝ | 1¾ | 1 15/16 | 6 | 6 1/32 | 5¼ | 5 7/32 | 5/32 | ⅛ | 7/8 | 8 | 8 | ¾×4¾ |
| 5 | 10¼ | 6 7/16 | 7 | 3/8 | 11/16 | 1 13/16 | 2 | 6½ | 6 17/32 | 5¾ | 5 23/32 | 5/32 | ⅛ | 7/8 | 8½ | 8 | ¾×4¾ |
| 6 | 11½ | 7⅝ | 8¼ | 7/16 | ¾ | 1⅞ | 2 1/16 | 7¾ | 7 25/32 | 7 | 6 31/32 | 5/32 | ⅛ | 7/8 | 9¾ | 8 | ¾×5 |
| 7 | 12½ | 8⅝ | 9 | 7/16 | 13/16 | 2⅛ | 2 5/16 | 8⅝ | 8 31/32 | 7⅞ | 7 27/32 | 5/32 | ⅛ | 1 | 10¾ | 8 | ⅞×5½ |
| 8 | 14¼ | 9¾ | 10¼ | ½ | ⅞ | 2¼ | 2½ | 9⅞ | 9 29/32 | 9 | 8 31/32 | 5/32 | ⅛ | 1 | 12 | 8 | ⅞×6 |
| 10 | 16½ | 12 | 12½ | 5/8 | 15/16 | 2⅜ | 2⅝ | 12 | 12 1/32 | 11 | 10 31/32 | 5/32 | ⅛ | 1× 1⅛ | 14¼ | 12 | ⅞×6¼ |
| 12 | 18¾ | 14⅝ | 14¾ | ¾ | 1 1/16 | 2½ | 2¾ | 14¼ | 14 9/32 | 13¼ | 13 7/32 | 5/32 | ⅛ | 1¼×1¼ | 16½ | 12 | 1 ×6½ |

## TABLE 24

### Flange Dimensions of "York" Tongue and Groove Round Flange Fittings



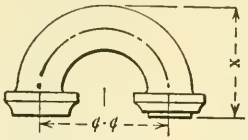




All Dimensions Given in Inches

| Size | $A$ | $B$ | $C$ | $D$ | $E$ | $F$ | $G$ | $H$ | $I$ | $J$ | $K$ | $L$ | $M$ | $N$ | $O$ | $P$ | $Q$ | $R$ | $S$ | $\frac{1}{16}$-in. Gasket | Bolts | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | | | | | | | | | | | | | | $B.C.$ | No. | Fitt. & Fitt. | Fitt. & Flge. |
| $2\frac{1}{2}$ | $7\frac{1}{4}$ | $2\frac{5}{8}$ | $4\frac{1}{4}$ | $\frac{5}{8}$ | $\frac{1}{2}$ | $1\frac{7}{16}$ | $1\frac{3}{8}$ | $3\frac{3}{4}$ | $3\frac{25}{32}$ | 3 | $2\frac{31}{32}$ | $\frac{5}{32}$ | $\frac{1}{8}$ | $\frac{13}{16}$ | $\frac{7}{8}$ | $2\frac{9}{16}$ | $\frac{9}{16}$ | $\frac{1}{2}$ | 3 | 3 × $3\frac{3}{4}$ | $5\frac{1}{2}$ | 6 | $\frac{3}{4}\times 3\frac{3}{4}$ | $\frac{3}{4}\times 4$ |
| 3 | $7\frac{7}{8}$ | $3\frac{1}{8}$ | $4\frac{3}{4}$ | $\frac{5}{8}$ | $\frac{9}{16}$ | $1\frac{9}{16}$ | $1\frac{1}{2}$ | $4\frac{3}{8}$ | $4\frac{13}{32}$ | $3\frac{5}{8}$ | $3\frac{19}{32}$ | $\frac{5}{32}$ | $\frac{1}{8}$ | $\frac{13}{16}$ | $\frac{7}{8}$ | $3\frac{1}{8}$ | $\frac{9}{16}$ | $\frac{1}{2}$ | $3\frac{5}{8}$ | $3\frac{5}{8}\times 4\frac{3}{8}$ | $6\frac{1}{8}$ | 6 | $\frac{3}{4}\times 4$ | $\frac{3}{4}\times 4\frac{1}{4}$ |
| $3\frac{1}{2}$ | $8\frac{5}{8}$ | $3\frac{5}{8}$ | $5\frac{3}{8}$ | $\frac{11}{16}$ | $\frac{9}{16}$ | $1\frac{9}{16}$ | $1\frac{1}{2}$ | $4\frac{7}{8}$ | $4\frac{29}{32}$ | $4\frac{1}{8}$ | $4\frac{3}{32}$ | $\frac{5}{32}$ | $\frac{1}{8}$ | $\frac{13}{16}$ | $\frac{7}{8}$ | $3\frac{11}{16}$ | $\frac{9}{16}$ | $\frac{1}{2}$ | $4\frac{1}{8}$ | $4\frac{1}{8}\times 4\frac{7}{8}$ | $6\frac{7}{8}$ | 8 | $\frac{3}{4}\times 4$ | $\frac{3}{4}\times 4\frac{1}{4}$ |
| 4 | $9\frac{3}{8}$ | $4\frac{1}{8}$ | $6\frac{1}{8}$ | $\frac{11}{16}$ | $\frac{5}{8}$ | $1\frac{11}{16}$ | $1\frac{5}{8}$ | $5\frac{1}{2}$ | $5\frac{17}{32}$ | $4\frac{3}{4}$ | $4\frac{23}{32}$ | $\frac{5}{32}$ | $\frac{1}{8}$ | $\frac{13}{16}$ | $\frac{7}{8}$ | $4\frac{3}{16}$ | $\frac{9}{16}$ | $\frac{1}{2}$ | $4\frac{3}{4}$ | $4\frac{3}{4}\times 5\frac{1}{2}$ | $7\frac{5}{8}$ | 8 | $\frac{3}{4}\times 4\frac{1}{4}$ | $\frac{3}{4}\times 4\frac{1}{2}$ |
| $4\frac{1}{2}$ | $9\frac{3}{4}$ | $4\frac{5}{8}$ | $6\frac{1}{2}$ | $\frac{3}{4}$ | $\frac{5}{8}$ | $1\frac{3}{4}$ | $1\frac{11}{16}$ | 6 | $6\frac{1}{32}$ | $5\frac{1}{4}$ | $5\frac{7}{32}$ | $\frac{5}{32}$ | $\frac{1}{8}$ | $\frac{13}{16}$ | $\frac{7}{8}$ | $4\frac{5}{8}$ | $\frac{9}{16}$ | $\frac{1}{2}$ | $5\frac{1}{4}$ | $5\frac{1}{4}\times 6$ | 8 | 8 | $\frac{3}{4}\times 4\frac{1}{2}$ | $\frac{3}{4}\times 4\frac{3}{4}$ |
| 5 | $10\frac{1}{4}$ | $5\frac{1}{8}$ | 7 | $\frac{3}{4}$ | $\frac{11}{16}$ | $1\frac{13}{16}$ | $1\frac{3}{4}$ | $6\frac{1}{2}$ | $6\frac{17}{32}$ | $5\frac{3}{4}$ | $5\frac{33}{32}$ | $\frac{5}{32}$ | $\frac{3}{4}$ | $\frac{13}{16}$ | $\frac{7}{8}$ | $5\frac{1}{4}$ | $\frac{9}{16}$ | $\frac{1}{2}$ | $5\frac{7}{8}$ | $5\frac{3}{4}\times 6\frac{1}{2}$ | $8\frac{1}{2}$ | 8 | $\frac{3}{4}\times 4\frac{1}{2}$ | $\frac{3}{4}\times 4\frac{3}{4}$ |
| 6 | $11\frac{1}{2}$ | $6\frac{1}{8}$ | $8\frac{1}{4}$ | $\frac{3}{4}$ | $\frac{3}{4}$ | $1\frac{15}{16}$ | $1\frac{7}{8}$ | $7\frac{3}{4}$ | $7\frac{25}{32}$ | 7 | $6\frac{31}{32}$ | $\frac{5}{32}$ | $\frac{3}{4}$ | $\frac{13}{16}$ | $\frac{7}{8}$ | $6\frac{1}{8}$ | $\frac{9}{16}$ | $\frac{1}{2}$ | $6\frac{7}{8}$ | $7\times 7\frac{3}{4}$ | $9\frac{3}{4}$ | 8 | $\frac{3}{4}\times 4\frac{3}{4}$ | $\frac{3}{4}\times 5$ |
| 7 | $12\frac{1}{2}$ | $7\frac{1}{8}$ | 9 | $\frac{3}{4}$ | $\frac{13}{16}$ | 2 | $1\frac{15}{16}$ | $8\frac{5}{8}$ | $8\frac{21}{32}$ | $7\frac{7}{8}$ | $7\frac{27}{32}$ | $\frac{5}{32}$ | $\frac{1}{8}$ | $\frac{15}{16}$ | 1 | $7\frac{1}{8}$ | $\frac{9}{16}$ | $\frac{1}{2}$ | 8 | $7\frac{7}{8}\times 8\frac{5}{8}$ | $10\frac{3}{4}$ | 8 | $\frac{7}{8}\times 5$ | $\frac{7}{8}\times 5\frac{1}{4}$ |
| 8 | $14\frac{1}{4}$ | $8\frac{1}{8}$ | $10\frac{1}{4}$ | $\frac{13}{16}$ | $\frac{7}{8}$ | $2\frac{1}{16}$ | 2 | $9\frac{7}{8}$ | $9\frac{29}{32}$ | 9 | $8\frac{31}{32}$ | $\frac{5}{32}$ | $\frac{1}{8}$ | $\frac{15}{16}$ | 1 | $8\frac{1}{8}$ | $\frac{9}{16}$ | $\frac{1}{2}$ | 9 | 9 × $9\frac{7}{8}$ | 12 | 8 | $\frac{7}{8}\times 5\frac{1}{4}$ | $\frac{7}{8}\times 5\frac{3}{4}$ |
| 10 | $16\frac{1}{2}$ | $10\frac{1}{8}$ | $12\frac{1}{2}$ | 1 | $\frac{15}{16}$ | $2\frac{1}{16}$ | 2 | 12 | $12\frac{1}{32}$ | 11 | $10\frac{31}{32}$ | $\frac{5}{32}$ | $\frac{1}{8}$ | $\frac{15}{16}$ | 1 | $10\frac{1}{8}$ | $\frac{9}{16}$ | $\frac{1}{2}$ | 11 | 11 ×12 | $14\frac{1}{4}$ | 12 | $\frac{7}{8}\times 5\frac{1}{4}$ | $\frac{7}{8}\times 5\frac{3}{4}$ |
| 12 | $18\frac{3}{4}$ | $12\frac{1}{8}$ | $14\frac{3}{4}$ | 1 | $1\frac{1}{16}$ | $2\frac{3}{16}$ | $2\frac{1}{8}$ | $14\frac{1}{4}$ | $14\frac{9}{32}$ | $13\frac{1}{4}$ | $13\frac{7}{32}$ | $\frac{5}{32}$ | $\frac{1}{8}$ | $1\frac{1}{8}$ | $1\frac{1}{4}$ | $12\frac{1}{8}$ | $\frac{9}{16}$ | $\frac{1}{2}$ | $13\frac{1}{8}$ | $13\frac{1}{4}\times 14\frac{1}{4}$ | $16\frac{1}{2}$ | 12 | 1 ×$5\frac{1}{2}$ | 1 ×6 |

## TABLE 25
### Length of "York" Return Bends
### Flanged Bends




All Dimensions Given in Inches

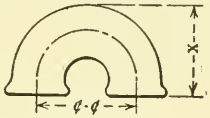
Oval Flanged

| Size | ¢–¢ | x | Size | ¢–¢ | x | Size | ¢–¢ | x |
|---|---|---|---|---|---|---|---|---|
| $1\frac{1}{4}$ | $3\frac{3}{4}$ | $4\frac{3}{16}$ | $1\frac{1}{4}$ | $4\frac{1}{2}$ | $4\frac{5}{8}$ | $1\frac{1}{4}$ | 6 | $5\frac{5}{8}$ |
| $1\frac{1}{4}$ | 4 | $4\frac{3}{8}$ | | | | | | |

Square Flanged

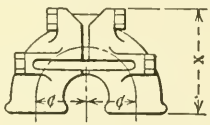
| Size | ¢–¢ | x | Size | ¢–¢ | x | Size | ¢–¢ | x |
|---|---|---|---|---|---|---|---|---|
| $1\frac{1}{4}$ | $4\frac{1}{8}$ | $4\frac{5}{8}$ | 2 | 8 | $6\frac{7}{8}$ | $2\frac{1}{2}$ | $5\frac{3}{4}$ | $4\frac{3}{4}$ |
| $1\frac{1}{4}$ | $8\frac{1}{2}$ | $6\frac{11}{16}$ | 2 | $8\frac{1}{2}$ | $7\frac{1}{8}$ | 3 | 12 | 10 |
| $1\frac{1}{2}$ | 6 | $5\frac{9}{16}$ | 2 | 9 | $7\frac{3}{8}$ | 4 | 12 | $10\frac{11}{16}$ |
| $1\frac{1}{2}$ | 8 | $6\frac{9}{16}$ | 2 | 10 | $7\frac{7}{8}$ | $1\frac{1}{4}$ | 4 | $4\frac{1}{2}$ |
| 2 | $4\frac{1}{8}$ | $4\frac{5}{8}$ | 2 | 12 | $8\frac{15}{16}$ | $1\frac{1}{4}$ | 6 | $5\frac{1}{4}$ |
| 2 | 6 | $5\frac{7}{8}$ | 2 | 15 | $5\frac{1}{4}$ | | | |

### Solid Screw Bends




| Size | ¢–¢ | x | Size | ¢–¢ | x | Size | ¢–¢ | x |
|---|---|---|---|---|---|---|---|---|
| 1 | $1\frac{3}{4}$ | $2\frac{9}{16}$ | $1\frac{1}{4}$ | 7 | $5\frac{9}{16}$ | 2 | $3\frac{1}{2}$ | $4\frac{1}{8}$ |
| 1 | $2\frac{1}{2}$ | $2\frac{15}{16}$ | $1\frac{1}{4}$ | 8 | $5\frac{15}{16}$ | 2 | $3\frac{3}{4}$ | $4\frac{3}{8}$ |
| 1 | 3 | $3\frac{1}{16}$ | $1\frac{1}{2}$ | 11 | $7\frac{9}{16}$ | 2 | 4 | $4\frac{1}{2}$ |
| 1 | 4 | $3\frac{13}{16}$ | $1\frac{1}{2}$ | $2\frac{9}{32}$ | $3\frac{1}{4}$ | 2 | $4\frac{3}{4}$ | $4\frac{7}{8}$ |
| 1 | 6 | $4\frac{13}{16}$ | $1\frac{1}{2}$ | $2\frac{3}{4}$ | $3\frac{11}{16}$ | 2 | 6 | $5\frac{9}{16}$ |
| $1\frac{1}{4}$ | $2\frac{1}{4}$ | $3\frac{5}{16}$ | $1\frac{1}{2}$ | $2\frac{3}{4}$ | $4\frac{1}{16}$ | 2 | 8 | $6\frac{1}{2}$ |
| $1\frac{1}{4}$ | $2\frac{1}{2}$ | $3\frac{1}{2}$ | $1\frac{1}{2}$ | 3 | $3\frac{13}{16}$ | 2 | 10 | $7\frac{1}{2}$ |
| $1\frac{1}{4}$ | 3 | $3\frac{1}{2}$ | $1\frac{1}{2}$ | 4 | $4\frac{3}{16}$ | $2\frac{1}{2}$ | 6 | $6\frac{1}{16}$ |
| $1\frac{1}{4}$ | $3\frac{1}{2}$ | $3\frac{11}{16}$ | $1\frac{1}{2}$ | 6 | $5\frac{9}{16}$ | 3 | 6 | $6\frac{5}{8}$ |
| $1\frac{1}{4}$ | 4 | $3\frac{15}{16}$ | $1\frac{1}{2}$ | 8 | $6\frac{3}{16}$ | $3\frac{1}{2}$ | 6 | $6\frac{5}{8}$ |
| $1\frac{1}{4}$ | $4\frac{1}{2}$ | $4\frac{7}{16}$ | 2 | 3 | $4\frac{5}{8}$ | 4 | 8 | $8\frac{1}{4}$ |
| $1\frac{1}{4}$ | 5 | $4\frac{11}{16}$ | 2 | 3 | $3\frac{15}{16}$ | 4 | 10 | $9\frac{5}{16}$ |
| $1\frac{1}{4}$ | 6 | $4\frac{15}{16}$ | 2 | $3\frac{1}{4}$ | $4\frac{1}{16}$ | 4 | 12 | $10\frac{1}{4}$ |
| $1\frac{1}{4}$ | $6\frac{3}{4}$ | $5\frac{9}{16}$ | | | | | | |

### Split Screw Bends

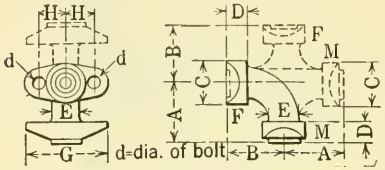



| Size | ¢–¢ | x | Size | ¢–¢ | x | Size | ¢–¢ | x |
|---|---|---|---|---|---|---|---|---|
| 1 | $2\frac{1}{2}$ | 3 | $1\frac{1}{4}$ | 5 | $5\frac{5}{8}$ | 2 | 4 | $5\frac{9}{16}$ |
| 1 | 3 | $3\frac{1}{4}$ | $1\frac{1}{4}$ | 6 | $6\frac{1}{8}$ | 2 | $4\frac{3}{4}$ | $5\frac{15}{16}$ |
| 1 | 4 | $3\frac{3}{4}$ | $1\frac{1}{2}$ | 4 | $5\frac{1}{8}$ | 2 | 6 | $6\frac{9}{16}$ |
| 1 | 6 | $4\frac{3}{4}$ | $1\frac{1}{2}$ | 6 | $6\frac{1}{8}$ | 2 | 8 | $7\frac{9}{16}$ |
| $1\frac{1}{4}$ | $2\frac{1}{2}$ | $4\frac{1}{4}$ | 2 | $3\frac{1}{4}$ | $5\frac{3}{16}$ | $2\frac{1}{2}$ | 6 | $7\frac{1}{8}$ |
| $1\frac{1}{4}$ | 3 | $4\frac{5}{8}$ | 2 | $3\frac{1}{2}$ | $5\frac{3}{16}$ | 3 | 6 | $7\frac{7}{16}$ |
| $1\frac{1}{4}$ | 4 | $5\frac{5}{8}$ | 2 | $3\frac{3}{4}$ | $5\frac{9}{16}$ | 4 | 8 | $9\frac{15}{16}$ |
| $1\frac{1}{4}$ | $4\frac{1}{2}$ | $5\frac{3}{16}$ | | | | | | |

## TABLE 26

### General Dimensions of "York" Ells, Crosses and Tees

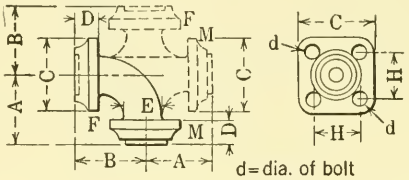
#### Oval Flange




All Dimensions Given in Inches

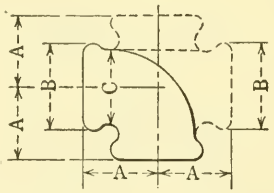
| Size | A | B | C | D | E | G | H | d |
|---|---|---|---|---|---|---|---|---|
| $\frac{1}{4}$ | $2\frac{7}{16}$ | $2\frac{3}{8}$ | $1\frac{3}{4}$ | $\frac{7}{8}$ | 1 | $3\frac{1}{2}$ | $1\frac{1}{8}$ | $\frac{1}{2}$ |
| $\frac{3}{8}$ and $\frac{1}{2}$ | $2\frac{13}{16}$ | $2\frac{3}{4}$ | $2\frac{1}{8}$ | 1 | $1\frac{1}{4}$ | $3\frac{3}{4}$ | $1\frac{5}{16}$ | $\frac{1}{2}$ |
| $\frac{3}{4}$ | $3\frac{5}{16}$ | $3\frac{1}{4}$ | $2\frac{3}{16}$ | $1\frac{1}{8}$ | $1\frac{3}{8}$ | $4\frac{1}{8}$ | $1\frac{3}{8}$ | $\frac{5}{8}$ |
| 1 | $3\frac{9}{16}$ | $3\frac{1}{2}$ | $2\frac{1}{2}$ | $1\frac{1}{4}$ | $1\frac{5}{8}$ | $4\frac{1}{2}$ | $1\frac{9}{16}$ | $\frac{5}{8}$ |
| $1\frac{1}{4}$ | $3\frac{13}{16}$ | $3\frac{3}{4}$ | $2\frac{7}{8}$ | $1\frac{3}{8}$ | $1\frac{7}{8}$ | $4\frac{7}{8}$ | $1\frac{21}{32}$ | $\frac{5}{8}$ |

#### Square Flange




| Size | A | B | C | D | E | H | d |
|---|---|---|---|---|---|---|---|
| $\frac{1}{2}$ | $2\frac{13}{16}$ | $2\frac{3}{4}$ | 3 | 1 | $1\frac{1}{4}$ | $1\frac{7}{8}$ | $\frac{1}{2}$ |
| 1 | $3\frac{9}{16}$ | $3\frac{1}{2}$ | $3\frac{5}{8}$ | $1\frac{1}{4}$ | $1\frac{5}{8}$ | $2\frac{1}{4}$ | $\frac{1}{2}$ |
| $1\frac{1}{4}$ | $3\frac{13}{16}$ | $3\frac{3}{4}$ | $3\frac{3}{4}$ | $1\frac{5}{16}$ | $1\frac{13}{16}$ | $2\frac{13}{32}$ | $\frac{5}{8}$ |
| $1\frac{1}{2}$ | $4\frac{5}{16}$ | $4\frac{1}{4}$ | 4 | $1\frac{7}{16}$ | $2\frac{1}{16}$ | $2\frac{5}{8}$ | $\frac{5}{8}$ |
| 2 | $4\frac{5}{16}$ | $4\frac{1}{2}$ | $4\frac{1}{2}$ | $1\frac{1}{2}$ | $2\frac{1}{4}$ | $3\frac{1}{18}$ | $\frac{5}{8}$ |
| $2\frac{1}{2}$ | $6\frac{1}{16}$ | 6 | $5\frac{5}{8}$ | $1\frac{7}{8}$ | $3\frac{5}{16}$ | 4 | $\frac{3}{4}$ |
| 3 | $6\frac{1}{16}$ | 6 | $6\frac{1}{8}$ | $1\frac{7}{8}$ | $3\frac{15}{16}$ | $4\frac{1}{2}$ | $\frac{3}{4}$ |
| $3\frac{1}{2}$ | $6\frac{9}{16}$ | $6\frac{1}{2}$ | $6\frac{3}{8}$ | $1\frac{7}{8}$ | $4\frac{1}{2}$ | $4\frac{9}{16}$ | $\frac{7}{8}$ |
| 4 | $7\frac{1}{16}$ | 7 | $7\frac{1}{8}$ | $1\frac{7}{8}$ | 5 | 5 | $\frac{7}{8}$ |
| 5 | $7\frac{9}{16}$ | $7\frac{1}{2}$ | 8 | 2 | $6\frac{1}{4}$ | $5\frac{7}{8}$ | $\frac{7}{8}$ |

#### Screw End

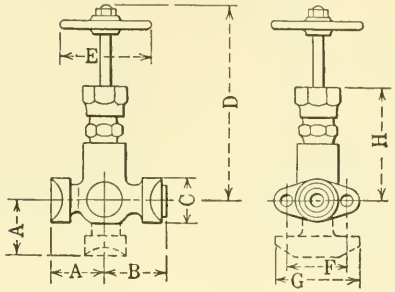



| Size | A | B | C | Size | A | B | C | Size | A | B | C |
|---|---|---|---|---|---|---|---|---|---|---|---|
| $\frac{1}{4}$ | 1 | $1\frac{1}{8}$ | 1 | $1\frac{1}{4}$ | $2\frac{1}{8}$ | $2\frac{21}{32}$ | $2\frac{5}{16}$ | $3\frac{1}{2}$ | 4 | $5\frac{3}{4}$ | $5\frac{1}{4}$ |
| $\frac{3}{8}$ | $1\frac{1}{8}$ | $1\frac{1}{4}$ | $1\frac{1}{8}$ | $1\frac{1}{2}$ | $2\frac{1}{2}$ | 3 | $2\frac{5}{8}$ | 4 | $4\frac{3}{8}$ | $6\frac{3}{8}$ | $5\frac{5}{8}$ |
| $\frac{1}{2}$ | $1\frac{5}{16}$ | $1\frac{15}{32}$ | $1\frac{5}{16}$ | 2 | $2\frac{7}{8}$ | $3\frac{3}{8}$ | $3\frac{1}{8}$ | 5 | $5\frac{1}{4}$ | $7\frac{1}{4}$ | $6\frac{3}{4}$ |
| $\frac{3}{4}$ | $1\frac{1}{2}$ | $1\frac{13}{16}$ | $1\frac{9}{16}$ | $2\frac{1}{2}$ | $3\frac{1}{4}$ | $4\frac{1}{4}$ | $3\frac{5}{8}$ | 6 | $6\frac{1}{4}$ | 9 | 8 |
| 1 | $1\frac{13}{16}$ | $2\frac{3}{16}$ | $1\frac{15}{16}$ | 3 | $3\frac{5}{8}$ | 5 | $4\frac{1}{2}$ | | | | |

## TABLE 27

### Principal Dimensions of "York" Ammonia Valves

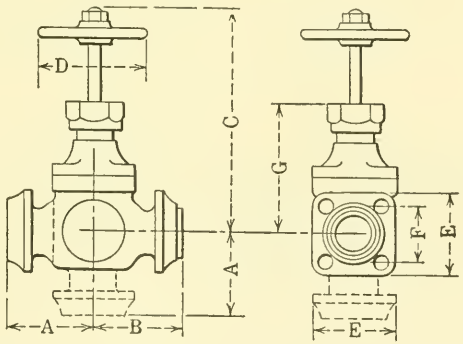
Oval Flange




All Dimensions Given in Inches

| Size | *A* | *B* | *C* | *D*, Closed | *D*, Open | *E* | *F* | *G* | *H* |
|---|---|---|---|---|---|---|---|---|---|
| $\frac{1}{4}$ | $2\frac{3}{8}$ | $2\frac{7}{16}$ | $1\frac{3}{4}$ | $7\frac{7}{8}$ | $8\frac{1}{4}$ | 4 | $2\frac{1}{4}$ | $3\frac{1}{2}$ | $4\frac{3}{8}$ |
| $\frac{3}{8}$ | $2\frac{3}{4}$ | $2\frac{13}{16}$ | $2\frac{1}{2}$ | $8\frac{5}{8}$ | $9\frac{1}{4}$ | 4 | $2\frac{5}{8}$ | $3\frac{3}{4}$ | 5 |
| $\frac{1}{2}$ | $2\frac{3}{4}$ | $2\frac{13}{16}$ | $2\frac{1}{2}$ | $8\frac{5}{8}$ | $9\frac{1}{8}$ | 4 | $2\frac{5}{8}$ | $3\frac{3}{4}$ | 5 |
| $\frac{3}{4}$ | $3\frac{1}{4}$ | $3\frac{5}{16}$ | $2\frac{3}{16}$ | 9 | $9\frac{11}{16}$ | 4 | $2\frac{3}{4}$ | $4\frac{1}{8}$ | $5\frac{11}{16}$ |
| 1 | $3\frac{1}{2}$ | $3\frac{9}{16}$ | $2\frac{1}{2}$ | $9\frac{1}{2}$ | $10\frac{1}{4}$ | 4 | $3\frac{1}{8}$ | $4\frac{1}{2}$ | $5\frac{7}{8}$ |
| $1\frac{1}{4}$ | $3\frac{3}{4}$ | $3\frac{13}{18}$ | $2\frac{3}{8}$ | $10\frac{13}{16}$ | $11\frac{1}{4}$ | $5\frac{1}{2}$ | $3\frac{5}{16}$ | $4\frac{7}{8}$ | $6\frac{15}{16}$ |

Square Flange

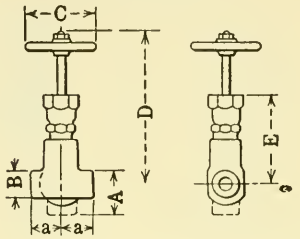



| Size | *A* | *B* | *C*, Closed | *C*, Open | *D* | *E* | *F* | *G* |
|---|---|---|---|---|---|---|---|---|
| $1\frac{1}{4}$ | $3\frac{3}{4}$ | $3\frac{13}{16}$ | $8\frac{9}{16}$ | $9\frac{1}{16}$ | $4\frac{3}{4}$ | $3\frac{3}{4}$ | $2\frac{3}{8}$ | $6\frac{7}{16}$ |
| $1\frac{1}{2}$ | $4\frac{1}{4}$ | $4\frac{5}{16}$ | $10\frac{13}{16}$ | $11\frac{1}{4}$ | $5\frac{1}{2}$ | 4 | $2\frac{5}{8}$ | $6\frac{15}{16}$ |
| 2 | $4\frac{1}{4}$ | $4\frac{5}{16}$ | $12\frac{1}{2}$ | $13\frac{1}{16}$ | $6\frac{3}{4}$ | $4\frac{1}{2}$ | $3\frac{1}{16}$ | $7\frac{3}{16}$ |
| $2\frac{1}{2}$ | 6 | $6\frac{1}{16}$ | $13\frac{5}{32}$ | $13\frac{13}{18}$ | $6\frac{3}{4}$ | $5\frac{5}{8}$ | 4 | $7\frac{13}{16}$ |
| 3 | 6 | $6\frac{1}{16}$ | $14\frac{5}{16}$ | $15\frac{5}{16}$ | 8 | $6\frac{1}{8}$ | $4\frac{1}{8}$ | $8\frac{15}{16}$ |
| $3\frac{1}{2}$ | $6\frac{1}{2}$ | $6\frac{9}{16}$ | $14\frac{7}{8}$ | $15\frac{7}{8}$ | 8 | $6\frac{5}{8}$ | $4\frac{9}{16}$ | $9\frac{1}{16}$ |
| 4 | 7 | $7\frac{1}{16}$ | $16\frac{1}{4}$ | $17\frac{3}{8}$ | 10 | $7\frac{1}{8}$ | 5 | $9\frac{3}{4}$ |
| 5 | $7\frac{1}{2}$ | $7\frac{9}{16}$ | 18 | $19\frac{3}{8}$ | 12 | 8 | $5\frac{1}{8}$ | $11\frac{1}{4}$ |

## TABLE 28

### Principal Dimensions of "York" Ammonia Valves

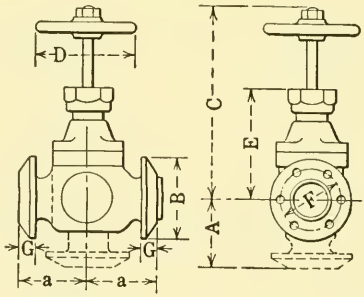
### Screw Valves




All Dimensions Given in Inches

| Size | *A* | *B* | *C* | *D*, Closed | *D*, Open | *E* |
|---|---|---|---|---|---|---|
| $\frac{1}{4}$ | $1\frac{3}{8}$ | $1\frac{3}{8}$ | 4 | $7\frac{7}{8}$ | $8\frac{1}{4}$ | $4\frac{5}{8}$ |
| $\frac{3}{8}$ | 2 | $1\frac{13}{16}$ | 4 | $8\frac{5}{8}$ | $9\frac{1}{8}$ | 5 |
| $\frac{1}{2}$ | 2 | $1\frac{13}{16}$ | 4 | $8\frac{5}{8}$ | $9\frac{1}{8}$ | 5 |
| $\frac{3}{4}$ | $2\frac{1}{8}$ | $2\frac{1}{16}$ | 4 | 9 | $9\frac{11}{16}$ | $5\frac{11}{16}$ |
| 1 | $2\frac{1}{4}$ | $2\frac{1}{4}$ | 4 | $9\frac{1}{2}$ | $10\frac{1}{4}$ | $5\frac{7}{8}$ |
| $1\frac{1}{4}$ | $2\frac{7}{8}$ | $2\frac{3}{4}$ | $5\frac{1}{2}$ | $10\frac{13}{16}$ | $11\frac{1}{4}$ | $6\frac{13}{16}$ |
| $1\frac{1}{2}$ | 3 | 3 | $5\frac{1}{2}$ | $10\frac{13}{16}$ | $11\frac{1}{4}$ | $6\frac{13}{16}$ |
| 2 | $3\frac{1}{2}$ | $3\frac{5}{8}$ | $6\frac{3}{4}$ | $12\frac{1}{2}$ | $13\frac{1}{16}$ | $7\frac{3}{16}$ |
| $2\frac{1}{2}$ | $4\frac{1}{4}$ | $4\frac{3}{8}$ | $6\frac{3}{4}$ | $13\frac{5}{32}$ | $13\frac{7}{16}$ | $7\frac{13}{16}$ |
| 3 | $4\frac{2}{4}$ | 5 | 8 | $14\frac{5}{16}$ | $15\frac{5}{16}$ | $8\frac{15}{16}$ |

### Round Flange Valves

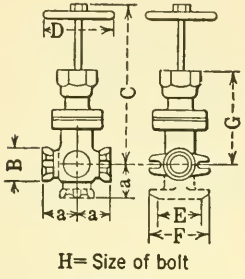


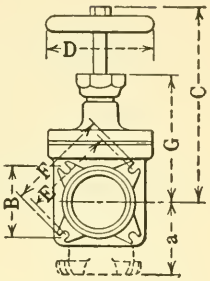

| Size | *A* | *B* | *C*, Closed | *C*, Open | *D* | *E* | *F* | *G* | Bolts |
|---|---|---|---|---|---|---|---|---|---|
| $2\frac{1}{2}$ | 6 | $7\frac{1}{8}$ | $15\frac{1}{8}$ | $15\frac{7}{8}$ | $6\frac{1}{4}$ | 10 | $5\frac{1}{2}$ | $1\frac{3}{8}$ | 6–$\frac{3}{4}$ |
| 3 | 6 | $7\frac{7}{8}$ | $14\frac{5}{16}$ | $15\frac{5}{16}$ | 8 | $8\frac{15}{16}$ | $6\frac{1}{2}$ | $1\frac{1}{2}$ | 6–$\frac{3}{4}$ |
| $3\frac{1}{2}$ | $6\frac{1}{2}$ | $8\frac{5}{8}$ | $14\frac{7}{8}$ | $15\frac{7}{8}$ | 8 | $9\frac{7}{16}$ | $6\frac{7}{8}$ | $1\frac{1}{2}$ | 8–$\frac{3}{4}$ |
| 4 | 7 | $9\frac{3}{8}$ | $16\frac{1}{4}$ | $17\frac{3}{8}$ | 10 | $9\frac{3}{4}$ | $7\frac{5}{8}$ | $1\frac{5}{8}$ | 8–$\frac{3}{4}$ |
| $4\frac{1}{2}$ | $7\frac{1}{2}$ | $9\frac{3}{4}$ | ..... | ..... | ..... | ..... | 8 | $1\frac{11}{16}$ | 8–$\frac{3}{4}$ |
| 5 | $7\frac{1}{2}$ | $10\frac{1}{4}$ | 18 | $19\frac{3}{8}$ | 12 | $11\frac{1}{4}$ | $8\frac{1}{2}$ | $1\frac{3}{4}$ | 8–$\frac{3}{4}$ |
| 6 | $8\frac{1}{2}$ | $11\frac{1}{2}$ | $20\frac{1}{16}$ | $21\frac{11}{16}$ | 14 | $13\frac{5}{16}$ | $9\frac{3}{4}$ | $1\frac{7}{8}$ | 8–$\frac{3}{4}$ |
| 7 | 10 | $12\frac{1}{2}$ | 21 | $22\frac{3}{4}$ | 14 | $14\frac{1}{4}$ | $10\frac{3}{4}$ | $1\frac{15}{16}$ | 8–$\frac{7}{8}$ |
| 8 | 12 | $14\frac{1}{4}$ | $22\frac{3}{8}$ | $24\frac{1}{2}$ | 14 | $16\frac{7}{16}$ | 12 | 2 | 8–$\frac{7}{8}$ |
| 10 | 14 | $16\frac{1}{2}$ | $24\frac{13}{16}$ | $27\frac{3}{16}$ | 18 | 18 | $14\frac{1}{4}$ | 2 | 12–$\frac{7}{8}$ |
| 12 | 16 | $18\frac{3}{4}$ | $27\frac{11}{16}$ | $30\frac{15}{16}$ | 18 | $20\frac{9}{16}$ | $16\frac{1}{2}$ | $2\frac{1}{8}$ | 12–1 |

## TABLE 29

### General Dimensions of "York" Gland End Fittings

### Globe and Angle Valves

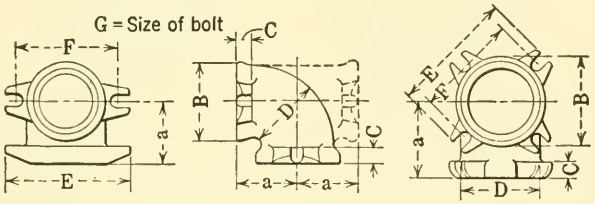



Dimensions below Heavy Line Indicate Four Bolt Fittings
All Dimensions Given in Inches

| Size | *A* | *B* | *C*, Closed | *C*, Open | *D* | *E* | *F* | *G* | *H* |
|---|---|---|---|---|---|---|---|---|---|
| ⅜ | 2¼ | 1¼ | 8⅝ | 9⅛ | 4 | 2⅛ | 3⅛ | 5 | ½ |
| ¾ | 2⅜ | 2⅛ | 9 | 9 11/16 | 4 | 2⅞ | 3⅞ | 5 11/16 | ½ |
| 1 | 2½ | 2⅜ | 9½ | 10¼ | 4 | 3⅛ | 4⅛ | 5⅞ | ½ |
| 1¼ | 3⅛ | 2⅞ | 10 13/16 | 11¼ | 5½ | 3½ | 4½ | 6 15/16 | ½ |
| 1½ | 3¼ | 3⅛ | 10 13/16 | 11¼ | 5½ | 3⅝ | 4⅝ | 6 13/16 | ½ |
| 2 | 3¾ | 3⅝ | 12½ | 13 1/16 | 6¾ | 4¼ | 5⅛ | 7 3/16 | ½ |
| 2½ | 4½ | 4¼ | 13 5/16 | 13 13/16 | 6¾ | 5 | 6⅛ | 7 13/16 | ⅝ |
| 3 | 5 | 5 | 14 5/16 | 15 5/16 | 8 | 6¼ | 7⅜ | 8 15/16 | ¾ |
| 3½ | 5½ | 5½ | 14⅞ | 15⅞ | 8 | 6⅝ | 7⅝ | 9 7/16 | ⅝ |
| 4 | 6 | 6¼ | 16¼ | 17⅜ | 10 | 7¼ | 8¾ | 9¾ | ¾ |
| 5 | 7 | 7½ | 18 | 19⅜ | 12 | 8½ | 10 | 11¼ | ¾ |

### Tees and Ells

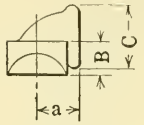


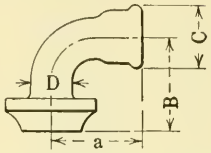

| Size | *A* | *B* | *C* | *D* | *E* | *F* | *G* |
|---|---|---|---|---|---|---|---|
| ½ | 1⅝ | 1¾ | ⅝ | 1½ | 3½ | 2½ | ½ |
| ¾ | 1¾ | 2⅛ | ¾ | 1 13/16 | 3⅞ | 2⅞ | ½ |
| 1 | 1⅞ | 2⅜ | ¾ | 2 1/16 | 4⅛ | 3⅛ | ½ |
| 1¼ | 2⅛ | 2⅞ | ¾ | 2½ | 4½ | 3½ | ½ |
| 1½ | 2⅜ | 3⅛ | ⅞ | 2¾ | 4⅝ | 3⅝ | ½ |
| 2 | 2⅞ | 3⅝ | ⅞ | 3 3/16 | 5¼ | 4¼ | ½ |
| 2½ | 3½ | 4¼ | 1⅛ | 3¾ | 6¼ | 5 | ⅝ |
| 3 | 3¾ | 5 | 1⅛ | 4⅝ | 7¾ | 6¼ | ¾ |
| 3½ | 4¾ | 5½ | 1⅛ | 5 1/16 | 7⅝ | 6½ | ⅝ |
| 4 | 5¼ | 6¼ | 1¼ | 5⅝ | 8¾ | 7¼ | ¾ |
| 5 | 6 | 7½ | 1½ | 6⅞ | 10 | 8½ | ¾ |

TABLE 30

GENERAL DIMENSIONS OF "YORK" SCREW AND FLANGE ELLS AND TEES

Screw and Flange Ells

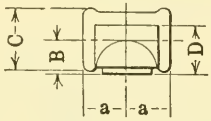


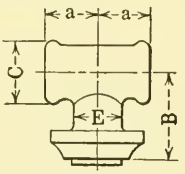




All Flange Dimensions Standard Except as Noted
Dimensions above Heavy Line Are for Oval, below for Square

| Size | *A* | *B* | *C* | *D* |
|---|---|---|---|---|
| $\frac{1}{4}$ | 1 | $\frac{3}{4}$ | $1\frac{1}{8}$ | |
| $\frac{3}{8}$ and $\frac{1}{2}$ | $1\frac{1}{8}$ | $\frac{7}{8}$ | $1\frac{5}{16}$ | |
| $\frac{3}{4}$ | $1\frac{1}{4}$ | 1 | $1\frac{13}{16}$ | |
| 1 | $1\frac{1}{2}$ | $1\frac{1}{4}$ | $2\frac{3}{16}$ | |
| $1\frac{1}{4}$ | $1\frac{3}{4}$ | $1\frac{3}{8}$ | $2\frac{21}{32}$ | |
| $\frac{1}{2}$ | 2 | $2\frac{3}{8}$ | $1\frac{9}{16}$ | $1\frac{1}{8}$ |
| 1 | $2\frac{3}{4}$ | 3 | $2\frac{3}{16}$ | $1\frac{1}{2}$ |
| $1\frac{1}{4}$ | $3\frac{3}{4}$ | $3\frac{3}{4}$ | $2\frac{21}{32}$ | $1\frac{13}{16}$ |
| $1\frac{1}{2}$ | $4\frac{1}{4}$ | $4\frac{1}{4}$ | 3 | $2\frac{3}{16}$ |
| 2 | $4\frac{1}{4}$ | $4\frac{1}{4}$ | $3\frac{1}{2}$ | $2\frac{3}{4}$ |
| $2\frac{1}{2}$ | 6 | 6 | $4\frac{1}{8}$ | $3\frac{5}{16}$ |
| 3 | 6 | 6 | $4\frac{7}{8}$ | $3\frac{15}{16}$ |
| $3\frac{1}{2}$ | $6\frac{1}{2}$ | $6\frac{1}{2}$ | $5\frac{5}{8}$ | $4\frac{1}{2}$ |
| 4 | 7 | 7 | $6\frac{3}{8}$ | 5 |
| 5 | $7\frac{1}{2}$ | $7\frac{1}{2}$ | $7\frac{3}{4}$ | $6\frac{1}{4}$ |

Screw and Flange Tees



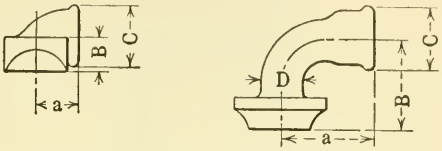




| Size | *A* | *B* | *C* | *D* | *E* |
|---|---|---|---|---|---|
| $\frac{1}{4}$ | 1 | $\frac{3}{4}$ | $1\frac{1}{8}$ | $\frac{7}{8}$ | |
| $\frac{3}{8}$ and $\frac{1}{2}$ | $1\frac{1}{8}$ | $\frac{15}{16}$ | $1\frac{15}{32}$ | $1\frac{1}{16}$ | |
| $\frac{3}{4}$ | $1\frac{1}{4}$ | $1\frac{1}{16}$ | $1\frac{13}{16}$ | $1\frac{7}{16}$ | |
| 1 | $1\frac{1}{2}$ | $1\frac{5}{16}$ | $2\frac{3}{16}$ | $1\frac{13}{16}$ | |
| $1\frac{1}{4}$ | $1\frac{3}{4}$ | $1\frac{7}{16}$ | $2\frac{21}{32}$ | $2\frac{1}{16}$ | |
| $\frac{1}{2}$ | | | | | |
| 1 | | | | | |
| $1\frac{1}{4}$ | $2\frac{1}{8}$ | $3\frac{13}{16}$ | $2\frac{21}{32}$ | ..... | $1\frac{13}{16}$ |
| $1\frac{1}{2}$ | $2\frac{1}{2}$ | $4\frac{5}{16}$ | 3 | ..... | $2\frac{3}{16}$ |
| 2 | $2\frac{7}{8}$ | $4\frac{5}{16}$ | $3\frac{5}{8}$ | ..... | $2\frac{3}{4}$ |
| $2\frac{1}{2}$ | $3\frac{1}{4}$ | $6\frac{1}{16}$ | $4\frac{1}{4}$ | ..... | $3\frac{5}{16}$ |
| 3 | $3\frac{5}{8}$ | $6\frac{1}{16}$ | 5 | ..... | $3\frac{15}{16}$ |
| $3\frac{1}{2}$ | 4 | $6\frac{9}{16}$ | $5\frac{3}{4}$ | ..... | $4\frac{1}{2}$ |
| 4 | $4\frac{3}{8}$ | $7\frac{1}{16}$ | $6\frac{3}{8}$ | ..... | 5 |
| 5 | $5\frac{1}{4}$ | $7\frac{9}{16}$ | $7\frac{3}{4}$ | ..... | $6\frac{1}{4}$ |

## TABLE 31

### General Dimensions of "York" Screw and Flange Ells and Tees

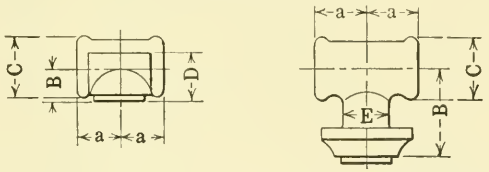
Screw and Flange Ells




All Flange Dimensions Standard Except as Noted
Dimensions above Heavy Line Are for Oval, below for Square

| Size | *A* | *B* | *C* | *D* |
|---|---|---|---|---|
| $\frac{1}{4}$ | 1 | $\frac{3}{4}$ | $1\frac{1}{8}$ | |
| $\frac{3}{8}$ and $\frac{1}{2}$ | $1\frac{1}{8}$ | $\frac{7}{8}$ | $1\frac{15}{32}$ | |
| $\frac{3}{4}$ | $1\frac{1}{4}$ | 1 | $1\frac{13}{16}$ | |
| 1 | $1\frac{1}{2}$ | $1\frac{1}{4}$ | $2\frac{3}{16}$ | |
| $1\frac{1}{4}$ | $1\frac{3}{4}$ | $1\frac{3}{8}$ | $2\frac{21}{32}$ | |
| $\frac{1}{2}$ | 2 | $2\frac{3}{8}$ | $1\frac{9}{16}$ | $1\frac{1}{8}$ |
| 1 | $2\frac{3}{4}$ | 3 | $2\frac{3}{16}$ | $1\frac{1}{2}$ |
| $1\frac{1}{4}$ | $3\frac{3}{4}$ | $3\frac{3}{4}$ | $2\frac{21}{32}$ | $1\frac{13}{16}$ |
| $1\frac{1}{2}$ | $4\frac{1}{4}$ | $4\frac{1}{4}$ | 3 | $2\frac{3}{16}$ |
| 2 | $4\frac{1}{4}$ | $4\frac{1}{4}$ | $3\frac{1}{2}$ | $2\frac{3}{4}$ |
| $2\frac{1}{2}$ | 6 | 6 | $4\frac{1}{8}$ | $3\frac{5}{16}$ |
| 3 | 6 | 6 | $4\frac{7}{8}$ | $3\frac{15}{16}$ |
| $3\frac{1}{2}$ | $6\frac{1}{2}$ | $6\frac{1}{2}$ | $5\frac{5}{8}$ | $4\frac{1}{2}$ |
| 4 | 7 | 7 | $6\frac{3}{8}$ | 5 |
| 5 | $7\frac{1}{2}$ | $7\frac{1}{2}$ | $7\frac{3}{4}$ | $6\frac{1}{4}$ |

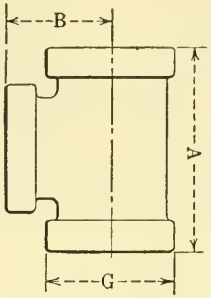
Screw and Flange Tees




| Size | *A* | *B* | *C* | *D* | *E* |
|---|---|---|---|---|---|
| $\frac{1}{4}$ | 1 | $\frac{3}{4}$ | $1\frac{1}{8}$ | $\frac{7}{8}$ | |
| $\frac{3}{8}$ and $\frac{1}{2}$ | $1\frac{1}{8}$ | $\frac{15}{16}$ | $1\frac{15}{32}$ | $1\frac{1}{16}$ | |
| $\frac{3}{4}$ | $1\frac{1}{8}$ | $1\frac{1}{16}$ | $1\frac{13}{16}$ | $1\frac{7}{16}$ | |
| 1 | $1\frac{1}{2}$ | $1\frac{5}{16}$ | $2\frac{3}{16}$ | $1\frac{13}{16}$ | |
| $1\frac{1}{4}$ | $1\frac{3}{4}$ | $1\frac{7}{16}$ | $2\frac{21}{32}$ | $2\frac{1}{16}$ | |
| $\frac{1}{2}$ | | | | | |
| 1 | | | | | |
| $1\frac{1}{4}$ | $2\frac{1}{8}$ | $3\frac{13}{16}$ | $2\frac{21}{32}$ | ..... | $1\frac{13}{16}$ |
| $1\frac{1}{2}$ | $2\frac{1}{2}$ | $4\frac{5}{16}$ | 3 | ..... | $2\frac{3}{16}$ |
| 2 | $2\frac{7}{8}$ | $4\frac{5}{16}$ | $3\frac{5}{8}$ | ..... | $2\frac{3}{4}$ |
| $2\frac{1}{2}$ | $3\frac{1}{4}$ | $6\frac{1}{16}$ | $4\frac{1}{4}$ | ..... | $3\frac{5}{16}$ |
| 3 | $3\frac{5}{8}$ | $6\frac{1}{16}$ | 5 | ..... | $3\frac{15}{16}$ |
| $3\frac{1}{2}$ | 4 | $6\frac{9}{16}$ | $5\frac{3}{4}$ | ..... | $4\frac{1}{2}$ |
| 4 | $4\frac{3}{8}$ | $7\frac{1}{16}$ | $6\frac{3}{8}$ | ..... | 5 |
| 5 | $5\frac{1}{4}$ | $7\frac{9}{16}$ | $7\frac{3}{4}$ | ..... | $6\frac{1}{4}$ |

TABLE 32

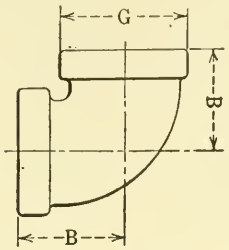
FRICK AMMONIA TEES—C. I. SCREW




| Size, Inches | A, Inches | B, Inches | G, Inches | Size, Inches | A, Inches | B, Inches | G, Inches |
|---|---|---|---|---|---|---|---|
| $\frac{1}{4}$ to $\frac{1}{4}\times\frac{1}{4}$ | $1\frac{3}{4}$ | $\frac{7}{8}$ | $1\frac{1}{8}$ | $\frac{1}{4}$ to $\frac{1}{4}\times1\frac{1}{4}$ | $4\frac{1}{4}$ | $2\frac{1}{8}$ | $2\frac{5}{8}$ |
| $\frac{3}{8}$ to $\frac{3}{8}\times\frac{3}{8}$ | 2 | 1 | $1\frac{1}{4}$ | $1\frac{1}{2}$ to $1\frac{1}{2}\times1\frac{1}{2}$ | 5 | $2\frac{1}{2}$ | 3 |
| $\frac{3}{8}$ to $\frac{1}{4}\times\frac{1}{4}$ | 2 | 1 | $1\frac{1}{4}$ | $1\frac{1}{2}$ to $\frac{1}{4}\times\frac{1}{4}$ | 5 | $2\frac{1}{2}$ | 3 |
| $\frac{1}{4}$ to $\frac{1}{4}\times\frac{3}{8}$ | 2 | 1 | $1\frac{1}{4}$ | $\frac{1}{4}$ to $\frac{1}{4}\times1\frac{1}{2}$ | 5 | $2\frac{1}{2}$ | 3 |
| $\frac{1}{2}$ to $\frac{1}{2}\times\frac{1}{2}$ | $2\frac{1}{2}$ | $1\frac{1}{4}$ | $1\frac{1}{2}$ | 2 to 2 $\times$2 | $5\frac{3}{4}$ | $2\frac{7}{8}$ | $3\frac{1}{2}$ |
| $\frac{1}{2}$ to $\frac{1}{4}\times\frac{1}{4}$ | $2\frac{1}{2}$ | $1\frac{1}{4}$ | $1\frac{1}{2}$ | 2 to $\frac{1}{4}\times\frac{1}{4}$ | $5\frac{3}{4}$ | $2\frac{7}{8}$ | $3\frac{1}{2}$ |
| $\frac{1}{4}$ to $\frac{1}{4}\times\frac{1}{2}$ | $2\frac{1}{2}$ | $1\frac{1}{4}$ | $1\frac{1}{2}$ | $\frac{1}{4}$ to $\frac{1}{4}\times$2 | $5\frac{3}{4}$ | $2\frac{7}{8}$ | $3\frac{1}{2}$ |
| $\frac{3}{4}$ to $\frac{3}{4}\times\frac{3}{4}$ | 3 | $1\frac{1}{2}$ | $1\frac{13}{16}$ | $2\frac{1}{2}$ to $2\frac{1}{2}\times2\frac{1}{2}$ | $10\frac{1}{2}$ | $5\frac{1}{4}$ | $4\frac{7}{8}$ |
| $\frac{3}{4}$ to $\frac{1}{4}\times\frac{1}{4}$ | 3 | $1\frac{1}{2}$ | $1\frac{13}{16}$ | $2\frac{1}{2}$ to 2 $\times$2 | $10\frac{1}{2}$ | $5\frac{1}{4}$ | $4\frac{7}{8}$ |
| $\frac{1}{4}$ to $\frac{1}{4}\times\frac{3}{4}$ | 3 | $1\frac{1}{2}$ | $1\frac{13}{16}$ | 3 to 3 $\times$3 | $10\frac{1}{2}$ | $5\frac{1}{4}$ | $5\frac{1}{2}$ |
| 1 to 1 $\times$1 | $3\frac{1}{2}$ | $1\frac{3}{4}$ | $2\frac{1}{8}$ | $3\frac{1}{2}$ to $3\frac{1}{2}\times3\frac{1}{2}$ | $11\frac{1}{2}$ | $5\frac{3}{4}$ | 6 |
| 1 to $\frac{1}{4}\times\frac{1}{4}$ | $3\frac{1}{2}$ | $1\frac{3}{4}$ | $2\frac{1}{8}$ | 4 to 4 $\times$4 | $12\frac{1}{2}$ | $6\frac{1}{4}$ | $6\frac{3}{4}$ |
| $\frac{1}{4}$ to $\frac{1}{4}\times$1 | $3\frac{1}{2}$ | $1\frac{3}{4}$ | $2\frac{1}{8}$ | $4\frac{1}{2}$ to $4\frac{1}{2}\times4\frac{1}{2}$ | $13\frac{1}{2}$ | $6\frac{3}{4}$ | $7\frac{3}{8}$ |
| $1\frac{1}{4}$ to $1\frac{1}{4}\times1\frac{1}{4}$ | $4\frac{1}{4}$ | $2\frac{1}{8}$ | $2\frac{5}{8}$ | 5 to 5 $\times$5 | $14\frac{1}{2}$ | $7\frac{1}{4}$ | 8 |
| $1\frac{1}{4}$ to $\frac{1}{4}\times\frac{1}{4}$ | $4\frac{1}{4}$ | $2\frac{1}{8}$ | $2\frac{5}{8}$ | | | | |

TABLE 32—*Continued*

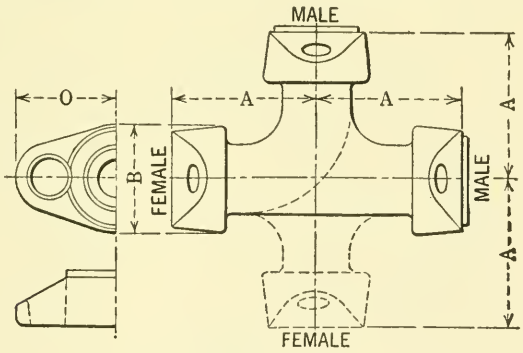
AMMONIA ELLS—C. I. SCREW




| Size, Inches | B, Inches | G, Inches | Size, Inches | B, Inches | G, Inches | Size, Inches | B, Inches | G, Inches |
|---|---|---|---|---|---|---|---|---|
| $\frac{1}{4}$ | $\frac{7}{8}$ | $1\frac{1}{8}$ | 1 | $1\frac{3}{4}$ | $2\frac{1}{8}$ | $2\times\frac{1}{4}$ | $2\frac{7}{8}$ | $3\frac{1}{2}$ |
| $\frac{3}{8}$ | 1 | $1\frac{1}{4}$ | $1\times\frac{1}{4}$ | $1\frac{3}{4}$ | $2\frac{1}{8}$ | $2\frac{1}{2}$ | $5\frac{1}{4}$ | $4\frac{7}{8}$ |
| $\frac{3}{8}\times\frac{1}{4}$ | 1 | $1\frac{1}{4}$ | $1\frac{1}{4}$ | $2\frac{1}{8}$ | $2\frac{5}{8}$ | 3 | $5\frac{1}{4}$ | $5\frac{1}{2}$ |
| $\frac{1}{2}$ | $1\frac{1}{4}$ | $1\frac{1}{2}$ | $1\frac{1}{4}\times\frac{1}{4}$ | $2\frac{1}{8}$ | $2\frac{5}{8}$ | $3\frac{1}{2}$ | $5\frac{3}{4}$ | 6 |
| $\frac{1}{2}\times\frac{1}{4}$ | $1\frac{1}{4}$ | $1\frac{1}{2}$ | $1\frac{1}{2}$ | $2\frac{1}{2}$ | 3 | 4 | $6\frac{1}{4}$ | $6\frac{3}{4}$ |
| $\frac{3}{4}$ | $1\frac{1}{2}$ | $1\frac{13}{16}$ | $1\frac{1}{2}\times\frac{1}{4}$ | $2\frac{1}{2}$ | 3 | $4\frac{1}{2}$ | | |
| $\frac{3}{4}\times\frac{1}{4}$ | $1\frac{1}{2}$ | $1\frac{13}{16}$ | 2 | $2\frac{7}{8}$ | $3\frac{1}{2}$ | 5 | $7\frac{1}{4}$ | 8 |

TABLE 33

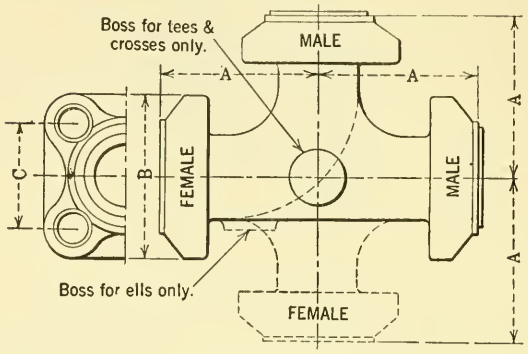
FRICK AMMONIA ELLS, TEES AND CROSSES—OVAL FLANGE




| Size, Inches | A, Inches | B, Inches | O, Inches | Size, Inches | A, Inches | B, Inches | O, Inches |
|---|---|---|---|---|---|---|---|
| $\frac{1}{4}$ | $2\frac{1}{8}$ | $1\frac{3}{4}$ | $1\frac{3}{4}$ | $\frac{3}{4}$ | 3 | $2\frac{3}{16}$ | $2\frac{1}{16}$ |
| $\frac{3}{8}$ | $2\frac{3}{8}$ | $2\frac{1}{16}$ | $1\frac{7}{8}$ | 1 | $3\frac{1}{4}$ | $2\frac{7}{16}$ | $2\frac{1}{4}$ |
| $\frac{1}{2}$ | $2\frac{3}{8}$ | $2\frac{1}{16}$ | $1\frac{7}{8}$ | | | | |

## TABLE 34

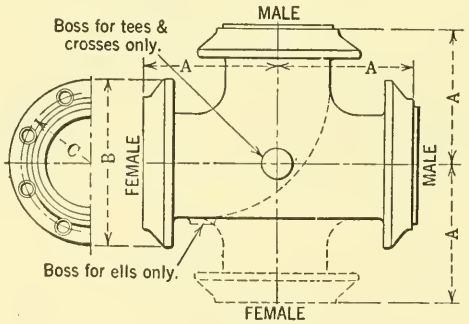
### Frick Ammonia Ells, Tees and Crosses—Square Flange




| Size, Inches | A, Inches | B, Inches | C, Inches | Bolts, Inches | Size, Inches | A, Inches | B, Inches | C, Inches | Bolts, Inches |
|---|---|---|---|---|---|---|---|---|---|
| $1\frac{1}{4}$ | $3\frac{3}{4}$ | $3\frac{3}{4}$ | $2\frac{3}{8}$ | $4–\frac{5}{8}\times2\frac{7}{8}$ | 3 | 6 | $6\frac{1}{8}$ | $4\frac{1}{8}$ | $4–\frac{3}{4}\times4\frac{1}{8}$ |
| $1\frac{1}{2}$ | $4\frac{1}{4}$ | 4 | $2\frac{5}{8}$ | $4–\frac{5}{8}\times3$ | $3\frac{1}{2}$ | $6\frac{1}{2}$ | $6\frac{5}{8}$ | $4\frac{9}{16}$ | $4–\frac{3}{4}\times4\frac{3}{8}$ |
| 2 | $4\frac{1}{4}$ | $4\frac{1}{2}$ | $3\frac{1}{16}$ | $4–\frac{5}{8}\times3\frac{1}{4}$ | 4 | 7 | $7\frac{1}{8}$ | 5 | $4–\frac{7}{8}\times4\frac{3}{4}$ |
| $2\frac{1}{2}$ | 6 | $5\frac{5}{8}$ | $3\frac{15}{16}$ | $4–\frac{3}{4}\times3\frac{7}{8}$ | | | | | |

## TABLE 35

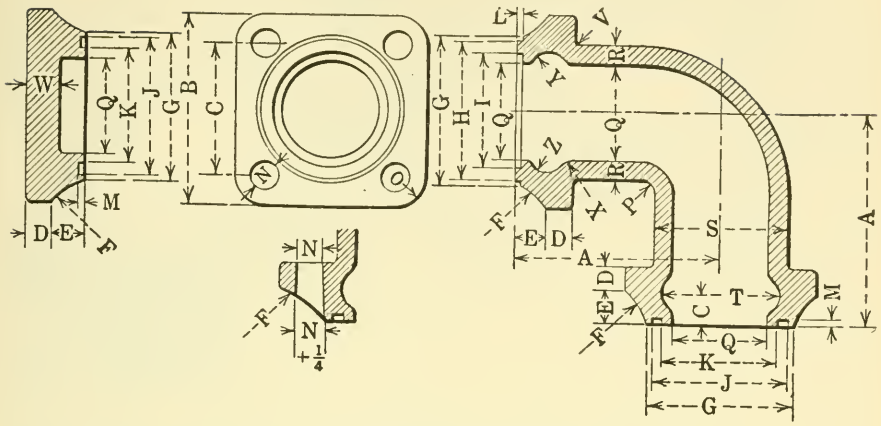
### Frick Ammonia Ells, Tees and Crosses—Round Flange




| Size, Inches | A, Inches | B, Inches | C, Inches | Bolts, Inches | Size, Inches | A, Inches | B, Inches | C, Inches | Bolts, Inches |
|---|---|---|---|---|---|---|---|---|---|
| 5 | $8\frac{1}{2}$ | $10\frac{1}{4}$ | $8\frac{1}{2}$ | $8–\frac{5}{8}\times4\frac{3}{4}$ | 8 | $12\frac{3}{4}$ | $14\frac{1}{2}$ | $12\frac{1}{2}$ | $8–\frac{7}{8}\times6$ |
| 6 | $9\frac{1}{2}$ | $11\frac{1}{2}$ | $9\frac{3}{4}$ | $8–\frac{3}{4}\times5\frac{1}{8}$ | 10 | 16 | 17 | 15 | $12–\frac{7}{8}\times5\frac{3}{4}$ |
| 7 | $11\frac{1}{2}$ | $13\frac{1}{2}$ | $11\frac{1}{8}$ | $8–\frac{7}{8}\times6$ | | | | | |

## TABLE 36

### Arctic Standard Square Flange Fittings




All Dimensions Given in Inches

| Pipe Size | A | B | C | D | E | F | G | H | I | Diameter of Bolt | Length of Bolt | Flange to Flange |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1/2 | 3 | 2 7/8 | 1 7/8 | 1/2 | 1/2 | 3 | 2 1/4 | 1 7/8 | 1 1/4 | 7/16 | 2 1/2 | 7/16 ×2 |
| 1 1/4 | 3 3/8 | 3 1/2 | 2 3/8 | 1/2 | 1/2 | 3 | 3 | 2 19/32 | 1 7/8 | 1/2 | 2 3/4 | 1/2 ×2 1/2 |
| 1 1/2 | 4 | 4 | 2 3/4 | 9/16 | 9/16 | 3 | 3 1/4 | 2 3/4 | 2 | 1/2 | 2 3/4 | |
| 2 | 4 3/4 | 4 1/2 | 3 1/16 | 5/8 | 5/8 | 3 | 3 3/4 | 3 1/4 | 2 1/2 | 5/8 | 3 1/4 | 5/8 ×3 |
| 2 1/2 | 6 | 5 1/2 | 3 5/8 | 5/8 | 5/8 | 3 | 4 1/8 | 3 25/32 | 3 | 5/8 | 4 1/4 | |
| 3 | 6 1/2 | 6 | 4 1/8 | 3/4 | 1 | 3 | 4 3/4 | 4 11/32 | 3 5/8 | 3/4 | 4 1/4 | |
| 3 1/2 | 7 | 6 1/2 | 4 1/2 | 3/4 | 1 | 3 | 5 1/2 | 4 7/8 | 4 3/32 | 3/4 | 4 1/2 | |
| 4 | 7 1/4 | 7 | 5 | 3/4 | 1 | 3 | 6 | 5 1/2 | 4 3/4 | 3/4 | 4 1/2 | |
| 4 1/2 | 7 1/2 | 7 1/2 | 5 1/2 | 3/4 | 1 | 3 | 6 5/8 | 6 | 5 1/4 | 3/4 | 4 1/2 | |
| 5 | 7 3/4 | 8 | 6 | 3/4 | 1 | 3 | 7 1/4 | 6 1/2 | 5 3/4 | 7/8 | 4 1/2 | |

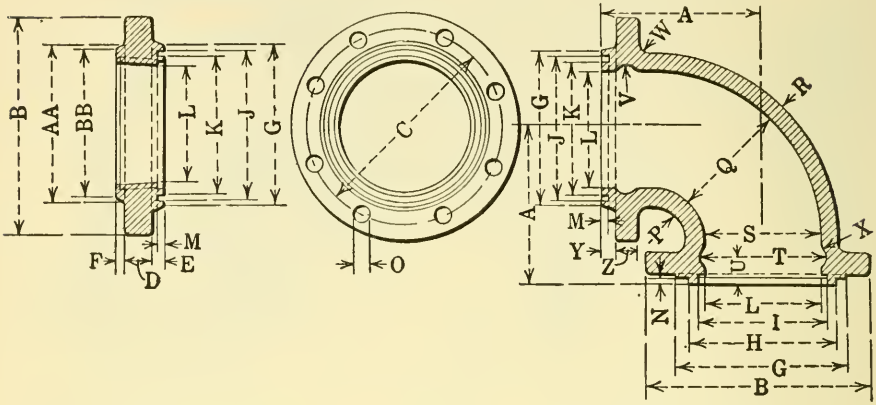
| Pipe Size | J | K | L | M | N | O | P | Q | R | Diameter of Bolt | Length of Bolt | Flange to Flange |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1/2 | 1 29/32 | 1 7/32 | 2/16 | 5/32 | 1/2 | 1/2 | 1/2 | 1 | 5/16 | 7/16 | 2 1/2 | 7/16 ×2 |
| 1 1/4 | 2 5/8 | 1 27/32 | 3/16 | 5/32 | 5/8 | 9/16 | 1/2 | 1 1/4 | 5/16 | 1/2 | 2 3/4 | 1/2 ×2 1/2 |
| 1 1/2 | 2 25/32 | 1 31/32 | 3/16 | 5/32 | 5/8 | 5/8 | 1/2 | 1 1/2 | 3/8 | 1/2 | 2 3/4 | |
| 2 | 3 9/32 | 2 15/32 | 3/16 | 5/32 | 3/4 | 23/32 | 1/2 | 2 | 7/16 | 5/8 | 3 1/4 | 5/8 ×3 |
| 2 1/2 | 3 13/16 | 2 31/32 | 3/16 | 5/32 | 7/8 | 13/16 | 1/2 | 2 1/2 | 1/2 | 3/4 | 4 1/4 | |
| 3 | 4 3/8 | 3 19/32 | 3/16 | 5/32 | 7/8 | 15/16 | 1/2 | 3 | 9/16 | 3/4 | 4 1/4 | |
| 3 1/2 | 4 29/32 | 4 1/16 | 3/16 | 5/32 | 7/8 | 1 | 1/2 | 3 1/2 | 9/16 | 3/4 | 4 1/2 | |
| 4 | 5 5/16 | 4 11/16 | 3/16 | 5/32 | 1 | 1 | 1/2 | 4 | 5/8 | 7/8 | 4 1/2 | |
| 4 1/2 | 6 1/16 | 5 5/16 | 3/16 | 5/32 | 1 | 1 | 1/2 | 4 1/2 | 5/8 | 7/8 | 4 1/2 | |
| 5 | 6 9/16 | 5 11/16 | 3/16 | 5/32 | 1 | 1 | 1/2 | 5 | 11/16 | 7/8 | 4 1/2 | |

| Pipe Size | S | T | U | V | W | X | Y | Z | Diameter of Bolt | Length of Bolt | Flange to Flange |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1/2 | 1 5/8 | 1 5/8 | 5/8 | 1/4 | . . . . . | 1/2 | 1/8 | 1 | 7/16 | 2 1/2 | 7/16 ×2 |
| 1 1/4 | 2 | 1 5/8 | 5/8 | 1/4 | . . . . . | 1/2 | 1/8 | 1 1/4 | 1/2 | 2 3/4 | 1/2 ×2 1/2 |
| 1 1/2 | 2 1/4 | 2 1/8 | 3/4 | 1/4 | . . . . . | 1/2 | 1/8 | 1 1/2 | 1/2 | 2 3/4 | |
| 2 | 2 7/8 | 2 1/2 | 3/4 | 1/4 | . . . . . | 1/2 | 1/8 | 2 | 5/8 | 3 1/4 | 5/8 ×3 |
| 2 1/2 | 3 1/2 | 3 1/8 | 1 | 1/4 | 1 | 5/8 | 1/8 | 2 1/2 | 3/4 | 4 1/4 | |
| 3 | 4 1/8 | 3 3/4 | 1 | 1/4 | 1 | 5/8 | 1/8 | 3 | 3/4 | 4 1/4 | |
| 3 1/2 | 4 5/8 | 4 1/8 | 1 | 1/4 | 1 1/8 | 5/8 | 1/8 | 3 1/2 | 3/4 | 4 1/2 | |
| 4 | 5 1/4 | 4 11/16 | 1 | 1/4 | 1 1/8 | 5/8 | 1/8 | 4 | 7/8 | 4 1/2 | |
| 4 1/2 | 5 3/4 | 5 3/16 | 1 | 1/4 | 1 3/16 | 5/8 | 1/8 | 4 1/2 | 7/8 | 4 1/2 | |
| 5 | 6 3/8 | 5 3/4 | 1 | 1/4 | 1 3/16 | 5/8 | 1/8 | 5 | 7/8 | 4 1/2 | |

Center to face dimension, "A," applies to ells, tees, crosses, angle valves, and straight-way valves.
Condenser inlet header tee branches are 7 in. center to face.
Condenser outlet header tee branches are 6 in. center to face.
Center to face of reducing tees is governed by dimensions for largest opening.
Fittings and valves are male one end, female other end. Branches on all tees and crosses are male, except on condenser outlet header tees.

## TABLE 37

### Arctic Standard Round Flanged Tongue and Groove Fittings

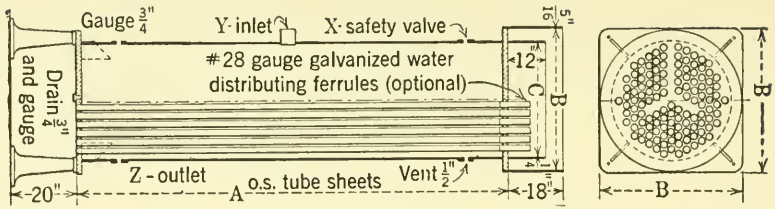



All Dimensions Except under Heading "Number of Bolts" Given in Inches

| Size of Pipe | Material of Flange | A | B | C | D | E | F | G | H | I | J | Number of Bolts | Diameter of Bolt | Length of Bolt | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | | | | | | | Fl. to fl. | Fl. to fit. | Fit. to fit. |
| 4½ | ST. | 7½ | 9¾ | 8 | ⅝ | ¾ | 7/16 | 6⅝ | 6 | 5¼ | 6 1/16 | 8 | ¾ | 3½ | 3¾ | 3¾ |
| 5 | ST. | 8 | 10¼ | 8½ | ⅝ | 13/16 | ½ | 7¼ | 6½ | 5¾ | 6 9/16 | 8 | ¾ | 3¾ | 3¾ | 4 |
| 6 | CI. | 8½ | 11⅜ | 9⅝ | 1⅛ | ½ | ⅜ | 8½ | 7¾ | 7 | 7 13/16 | 8 | ¾ | 4 | 4 | 4 |
| 7 | CI. | 9 | 12 7/8 | 10⅞ | 1 5/16 | 7/16 | ⅜ | 9⅝ | 8⅜ | 8 | 8 13/16 | 8 | ⅞ | 4½ | 4½ | 4½ |
| 8 | CI. | 9½ | 13⅝ | 11⅝ | 1⅜ | ½ | ⅜ | 10¼ | 9¾ | 9 | 9 13/16 | 12 | ¾ | 4½ | 4½ | 4½ |
| | | K | L | M | N | O | P | Q | R | S | T | | | | | |
| 4½ | ST. | 5 3/16 | 4⅝ | 5/32 | 3/16 | ⅞ | 1½ | 4½ | ⅝ | 5¾ | 5¼ | 8 | ¾ | 3½ | 3¾ | 3¾ |
| 5 | ST. | 5 11/16 | 5⅛ | 5/32 | 3/16 | ⅞ | 1½ | 5 | 11/16 | 6⅜ | 5¾ | 8 | ¾ | 3¾ | 3¾ | 4 |
| 6 | CI. | 6 1/16 | 6⅛ | 5/32 | 3/16 | ⅞ | 1½ | 6 | ¾ | 7½ | 6 13/16 | 8 | ¾ | 4 | 4 | 4 |
| 7 | CI. | 7 1/16 | 7⅛ | 5/32 | 3/16 | 1 | 1½ | 7 | 13/16 | 8⅝ | 8 | 8 | ⅞ | 4½ | 4½ | 4½ |
| 8 | CI. | 8 1/16 | 8⅛ | 5/32 | 3/16 | ⅞ | 2 | 8 | ⅞ | 9¾ | 8⅞ | 12 | ¾ | 4½ | 4½ | 4½ |
| | | U | V | W | X | Y | Z | AA | BB | | | | | | | |
| 4½ | ST. | ¾ | ⅜ | ¼ | ¾ | 5/16 | 1⅛ | 6 | 5⅞ | ... | ... | 8 | ¾ | 3½ | 3¾ | 3¾ |
| 5 | ST. | ⅞ | 7/16 | ¼ | 15/16 | ⅜ | 1 3/16 | 6½ | 6⅜ | ... | ... | 8 | ¾ | 3¾ | 3¾ | 4 |
| 6 | CI. | ⅞ | 7/16 | ¼ | 1 | ⅜ | 1¼ | 7⅝ | 7½ | ... | ... | 8 | ¾ | 4 | 4 | 4 |
| 7 | CI. | ⅞ | 7/16 | ¼ | 1 1/16 | 7/16 | 1 5/16 | 8¾ | 8⅝ | ... | ... | 8 | ⅞ | 4½ | 4½ | 4½ |
| 8 | CI. | 1 | ½ | ¼ | 1 1/16 | ½ | 1⅜ | 9¾ | 9⅝ | ... | ... | 12 | ¾ | 4½ | 4½ | 4½ |

## TABLE 38

### Struthers-Wells Co. Condensers

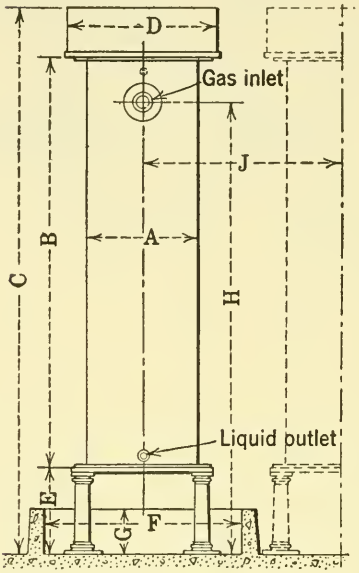



| Outside Diameter, Inches | Thickness of Shell, Inches | Thickness of Tube Sheet, Inches | Number of Tubes | Length of Tubes | Dimensions A | | B | C | S-V X | Inlet Y | Outlet Z | Tube Surface, Square Feet | Capacity of Refrigeration, Tons |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Ft. | Ft. | In. | Inches | Inches | Inches | Inches | Inches | | |
| 16 | 1/4 | 1 3/8 | 22 | 6 | 5 | 11 1/2 | 28 | 16 | 3/4 | 2 | 3/4 | 66 | 5.5 |
| | | | | 8 | 7 | 11 1/2 | | | | | | 88 | 7.3 |
| | | | | 10 | 9 | 11 1/2 | | | | | | 110 | 9.2 |
| 18 | 1/4 | 1 3/8 | 30 | 8 | 7 | 11 1/2 | 30 | 18 | 3/4 | 2 | 3/4 | 120 | 10.0 |
| | | | | 10 | 9 | 11 1/2 | | | | | | 150 | 12.5 |
| | | | | 12 | 11 | 11 1/2 | | | | | | 180 | 15.0 |
| 20 | 5/16 | 1 3/8 | 35 | 10 | 9 | 11 1/2 | 32 | 20 | 3/4 | 2 1/2 | 3/4 | 175 | 14.6 |
| | | | | 12 | 11 | 11 1/2 | | | | | | 210 | 17.5 |
| | | | | 14 | 13 | 11 1/2 | | | | | | 245 | 20.4 |
| 22 | 5/16 | 1 3/8 | 44 | 12 | 11 | 11 1/2 | 34 | 22 | 3/4 | 2 1/2 | 3/4 | 265 | 22.1 |
| | | | | 14 | 13 | 11 1/2 | | | | | | 308 | 25.6 |
| | | | | 16 | 15 | 11 1/2 | | | | | | 352 | 29.3 |
| 24 | 3/8 | 1 3/8 | 50 | 14 | 13 | 11 1/2 | 36 | 24 | 1 | 2 1/2 | 1 | 350 | 29.1 |
| | | | | 16 | 15 | 11 1/2 | | | | | | 400 | 33.3 |
| | | | | 18 | 17 | 11 1/2 | | | | | | 450 | 37.5 |
| 26 | 3/8 | 1 3/8 | 62 | 14 | 13 | 11 1/2 | 38 | 26 | 1 | 2 1/2 | 1 | 435 | 36.3 |
| | | | | 16 | 15 | 11 1/2 | | | | | | 495 | 41.2 |
| | | | | 18 | 17 | 11 1/2 | | | | | | 557 | 46.4 |

Note.—Tubes are 2 in. O. D. × No. 10 B. W. G. Charcoal Iron. Condenser electrically welded throughout unless otherwise specified. Test 500 lb. hydro. pressure.

TABLE 39

CARBONDALE CONDENSERS




| Refrigeration Capacity, Tons | A | B | | C | | D | E | F | |
|---|---|---|---|---|---|---|---|---|---|
| | Inches | Ft. | In. | Ft. | In. | Inches | Inches | Ft. | In. |
| 24 | $26\frac{1}{2}$ | 8 | $4\frac{1}{4}$ | 11 | 5 | $37\frac{1}{2}$ | $23\frac{3}{8}$ | 4 | 4 |
| 32 | $26\frac{1}{2}$ | 11 | $1\frac{1}{2}$ | 14 | $2\frac{1}{4}$ | $37\frac{1}{2}$ | $23\frac{3}{8}$ | 4 | 4 |
| 38 | $26\frac{1}{2}$ | 13 | $1\frac{1}{2}$ | 16 | $2\frac{1}{4}$ | $37\frac{1}{2}$ | $23\frac{3}{8}$ | 4 | 4 |
| 55 | 30 | 15 | $0\frac{1}{4}$ | 18 | 1 | 41 | $23\frac{3}{8}$ | 4 | 6 |
| 62 | 30 | 17 | $0\frac{1}{4}$ | 20 | 1 | 41 | $23\frac{3}{8}$ | 4 | 6 |
| 67 | $36\frac{3}{4}$ | 11 | $1\frac{1}{2}$ | 14 | $4\frac{1}{4}$ | $47\frac{3}{4}$ | $25\frac{3}{8}$ | 5 | 2 |
| 79 | $36\frac{3}{4}$ | 13 | $1\frac{1}{2}$ | 16 | $4\frac{1}{4}$ | $47\frac{3}{4}$ | $25\frac{3}{8}$ | 5 | 2 |
| 90 | $36\frac{3}{4}$ | 15 | $0\frac{1}{4}$ | 18 | 3 | $47\frac{3}{4}$ | $25\frac{1}{3}$ | 5 | 2 |
| 106 | $46\frac{1}{4}$ | 11 | $1\frac{1}{2}$ | 14 | $2\frac{1}{4}$ | $57\frac{3}{4}$ | $23\frac{3}{8}$ | 5 | 6 |
| 125 | $46\frac{1}{4}$ | 13 | $1\frac{1}{2}$ | 16 | $4\frac{1}{4}$ | $57\frac{3}{4}$ | $25\frac{3}{8}$ | 5 | 6 |
| | G | H | | J | | Gas | Liquor | Pump Out | |
| | Inches | Ft. | In. | Ft. | In. | Inches | Inches | Inches | |
| 24 | 12 | 8 | $9\frac{5}{8}$ | 3 | 8 | $1\frac{1}{2}$ | 1 | 1 | |
| 32 | 12 | 11 | $6\frac{7}{8}$ | 3 | 8 | 2 | $1\frac{1}{4}$ | 1 | |
| 38 | 12 | 13 | $6\frac{7}{8}$ | 3 | 8 | 2 | $1\frac{1}{4}$ | 1 | |
| 55 | 12 | 15 | $5\frac{5}{8}$ | 4 | 0 | $2\frac{1}{2}$ | $1\frac{1}{4}$ | 1 | |
| 62 | 12 | 17 | $5\frac{5}{8}$ | 4 | 0 | $2\frac{1}{2}$ | $1\frac{1}{4}$ | 1 | |
| 67 | 12 | 11 | $8\frac{7}{8}$ | 4 | 6 | $2\frac{1}{2}$ | $1\frac{1}{2}$ | 1 | |
| 79 | 12 | 13 | $8\frac{7}{8}$ | 4 | 6 | $2\frac{1}{2}$ | $1\frac{1}{2}$ | 1 | |
| 90 | 12 | 15 | $7\frac{5}{8}$ | 4 | 6 | 3 | 2 | 1 | |
| 106 | 12 | 11 | $6\frac{7}{8}$ | 5 | 6 | 3 | 2 | 1 | |
| 125 | 12 | 13 | $8\frac{1}{8}$ | 5 | 6 | 3 | 2 | $1\frac{1}{4}$ | |

TABLE 40

ATMOSPHERIC AMMONIA CONDENSERS 2-IN. PIPE—20 FT. LONG

| Pipes High | Gallons per Minute per Coil | Initial Temperature of Condensing Water | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 60 Deg. F. | | | 65 Deg. F. | | | 70 Deg. F. | | | 75 Deg. F. | | | 80 Deg. F. | | | 85 Deg. F. | | | 90 Deg. F. | | |
| | | Condenser pressure, pounds per square inch (gage) | | | | | | | | | | | | | | | | | | | | |
| | | 145 | 165 | 185 | 145 | 165 | 185 | 155 | 175 | 185 | 165 | 185 | 205 | 175 | 185 | 205 | 185 | 205 | 215 | 205 | 215 | 235 |
| | | Tons of refrigeration per 24 hours | | | | | | | | | | | | | | | | | | | | |
| 8 | 20 | 6 | 7 | 8 | 5 | 6 | 7 | 5 | 6 | 7 | 5 | 6 | 7 | 5 | 6 | 7 | 4 | 5 | 6 | 4 | 5 | 6 |
| | 15 | 5 | 6 | 7 | 4 | 5 | 6 | 4 | 5 | 6 | 4 | 5 | 6 | 4 | 5 | 6 | 3 | 4 | 5 | 3 | 4 | 5 |
| | 10 | 4 | 5 | 6 | 3 | 4 | 5 | 3 | 4 | 5 | 3 | 4 | 5 | 3 | 4 | 5 | 2 | 3 | 4 | 2 | 3 | 4 |
| 12 | 20 | 7 | 8 | 9 | 6 | 7 | 8 | 6 | 7 | 8 | 6 | 7 | 8 | 6 | 7 | 8 | 6 | 7 | 8 | 6 | 7 | 8 |
| | 15 | 6 | 7 | 8 | 5 | 6 | 7 | 5 | 6 | 7 | 5 | 6 | 7 | 5 | 6 | 7 | 5 | 6 | 7 | 5 | 6 | 7 |
| | 10 | 5 | 6 | 7 | 4 | 5 | 6 | 4 | 5 | 6 | 4 | 5 | 6 | 4 | 5 | 6 | 4 | 5 | 6 | 4 | 5 | 6 |
| 16 | 25 | 8 | 10 | 12 | 7 | 8 | 10 | 7 | 8 | 10 | 7 | 8 | 9 | 7 | 8 | 9 | 7 | 8 | 9 | 7 | 8 | 9 |
| | 20 | 7 | 9 | 11 | 6 | 7 | 9 | 6 | 7 | 9 | 6 | 7 | 8 | 6 | 7 | 8 | 6 | 7 | 8 | 6 | 7 | 8 |
| | 15 | 6 | 8 | 10 | 5 | 6 | 8 | 5 | 6 | 8 | 5 | 6 | 7 | 5 | 6 | 7 | 5 | 6 | 7 | 5 | 6 | 7 |
| 20 | 25 | 10 | 12 | 14 | 9 | 10 | 12 | 9 | 10 | 12 | 9 | 10 | 12 | 9 | 10 | 11 | 8 | 9 | 10 | 8 | 9 | 10 |
| | 20 | 9 | 11 | 13 | 8 | 9 | 11 | 8 | 9 | 11 | 8 | 9 | 10 | 8 | 9 | 10 | 7 | 8 | 9 | 7 | 8 | 9 |
| | 15 | 8 | 10 | 12 | 7 | 8 | 10 | 7 | 8 | 10 | 7 | 8 | 9 | 7 | 8 | 9 | 6 | 7 | 8 | 6 | 7 | 8 |
| 24 | 30 | 12 | 13 | 15 | 11 | 12 | 14 | 11 | 12 | 13 | 11 | 12 | 13 | 10 | 11 | 12 | 10 | 11 | 12 | 10 | 11 | 12 |
| | 25 | 11 | 12 | 14 | 10 | 11 | 12 | 10 | 11 | 12 | 10 | 11 | 12 | 9 | 10 | 11 | 9 | 10 | 11 | 9 | 10 | 11 |
| | 20 | 10 | 11 | 12 | 9 | 10 | 11 | 9 | 10 | 11 | 9 | 10 | 11 | 8 | 9 | 10 | 8 | 9 | 10 | 8 | 9 | 10 |

For condensers 10 ft. long divide above total G.P.M. and tons of refrigeration by 2.

## TABLE 41

CAPACITIES ATMOSPHERIC DRIP AMMONIA CONDENSERS 2-IN. PIPE—20 FT. LONG

| Pipes per Stand. | Gallons per Minute per Coil | Initial Temperature of Condensing Water | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 60 Deg. F. | | | 65 Deg. F. | | | 70 Deg. F. | | | 75 Deg. F. | | | 80 Deg. F. | | | 85 Deg. F. | | | 90 Deg. F. | | |
| | | Condenser pressure, pounds per square inch (gage) | | | | | | | | | | | | | | | | | | | | |
| | | 145 | 165 | 185 | 145 | 165 | 185 | 155 | 175 | 185 | 165 | 185 | 205 | 175 | 185 | 205 | 185 | 205 | 215 | 205 | 215 | 235 |
| | | Tons of refrigeration per 24 hours | | | | | | | | | | | | | | | | | | | | |
| 12 | 10 | 5.6 | 8.0 | 10.1 | 4.1 | 6.4 | 8.4 | 3.6 | 5.8 | 6.8 | 3.2 | 5.2 | 7.2 | 2.6 | 3.6 | 5.6 | 2.0 | 4.0 | 4.9 | 2.4 | 3.2 | 5.1 |
| | 15 | 8.5 | 12.0 | 15.1 | 6.1 | 9.5 | 12.7 | 5.4 | 8.7 | 10.3 | 4.8 | 7.9 | 10.8 | 3.9 | 5.4 | 8.4 | 3.0 | 6.0 | 7.4 | 3.6 | 4.8 | 7.6 |
| | 20 | 11.3 | 16.0 | 20.2 | 8.2 | 12.7 | 16.9 | 7.3 | 11.7 | 13.7 | 6.4 | 10.5 | 14.4 | 5.2 | 7.3 | 11.2 | 4.1 | 8.0 | 9.8 | 4.7 | 6.4 | 10.2 |
| | 25 | 14.1 | 20.0 | 24.8 | 10.2 | 15.9 | 21.2 | 9.1 | 14.6 | 17.2 | 8.0 | 13.1 | 18.0 | 6.5 | 9.1 | 14.0 | 5.1 | 10.0 | 12.3 | 6.0 | 8.0 | 12.7 |
| | 30 | 16.5 | 20.0 | .... | 12.2 | 17.9 | .... | 10.9 | 16.8 | 18.8 | 9.6 | 15.8 | 19.4 | 7.8 | 10.9 | 16.5 | 6.1 | 12.0 | 14.8 | 7.1 | 9.6 | 15.3 |
| | 35 | .... | .... | .... | 13.6 | .... | .... | 12.7 | .... | .... | 11.1 | .... | .... | 9.1 | 12.7 | .... | 7.1 | 13.4 | 15.2 | 8.3 | 12.1 | |
| | 40 | .... | .... | .... | .... | .... | .... | .... | .... | .... | 11.9 | .... | .... | 10.5 | .... | .... | 8.1 | .... | .... | 9.1 | | |
| 16 | 25 | 14.1 | 20.0 | 25.0 | 10.2 | 15.9 | 21.2 | 9.1 | 14.6 | 17.2 | 8.0 | 13.1 | 18.0 | 6.5 | 9.1 | 14.0 | 5.1 | 10.0 | 12.3 | 6.0 | 8.0 | 12.7 |
| | 30 | 16.8 | 23.9 | 30.0 | 12.2 | 18.9 | 25.3 | 10.9 | 17.3 | 20.5 | 9.6 | 15.8 | 21.5 | 7.8 | 10.9 | 16.5 | 6.1 | 12.0 | 14.8 | 7.1 | 9.6 | 15.3 |
| | 35 | 19.6 | 27.4 | 33.4 | 14.2 | 22.0 | 29.0 | 12.6 | 20.2 | 23.9 | 11.0 | 18.3 | 25.0 | 9.1 | 12.7 | 19.5 | 7.1 | 14.0 | 17.2 | 8.2 | 11.2 | 17.6 |
| | 40 | 22.2 | .... | .... | 16.2 | 23.8 | .... | 14.4 | 22.4 | 25.0 | 12.5 | 21.1 | 25.8 | 10.3 | 14.8 | 21.9 | 8.1 | 16.0 | 19.5 | 9.4 | 12.8 | 20.1 |
| | 45 | .... | .... | .... | 18.2 | .... | .... | 16.3 | .... | .... | 14.2 | .... | .... | 11.6 | 16.4 | .... | 9.1 | 18.0 | 20.2 | 10.6 | 14.4 | 20.6 |
| | 50 | .... | .... | .... | .... | .... | .... | 17.0 | .... | .... | 15.5 | .... | .... | 12.8 | 17.1 | .... | 10.1 | .... | .... | 11.8 | 16.0 | |

For 10-ft. condensers divide the above capacities and G.P.M. by 2.

TABLE 42

CAPACITIES DOUBLE-PIPE AMMONIA CONDENSERS $1\frac{1}{4}$ and 2-IN. PIPE—20 FT. LONG

| Pipe High | Gallons per Minute per Coil | Feet Head | Initial Temperature of Condensing Water | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | 60 Deg. F. | | | 65 Deg. F. | | | 70 Deg. F. | | | 75 Deg. F. | | | 80 Deg. F. | | | 85 Deg. F. | | | 90 Deg. F. | | |
| | | | Condenser pressure, pounds per square inch (gage) | | | | | | | | | | | | | | | | | | | | |
| | | | 145 | 165 | 185 | 165 | 185 | 195 | 175 | 185 | 205 | 185 | 205 | 215 | 185 | 205 | 215 | 185 | 205 | 215 | 205 | 215 | 235 |
| | | | Tons of refrigeration per 24 hours | | | | | | | | | | | | | | | | | | | | |
| | 5 | 1 | 2.0 | 2.5 | 3.0 | 2.0 | 2.5 | 2.5 | 1.5 | 2.0 | 2.5 | 1.5 | 1.8 | 2.0 | 1.3 | 1.5 | 2.0 | 1.0 | 1.2 | 1.3 | 1.0 | 1.0 | 1.5 |
| 4 | 10 | 3 | 2.5 | 4.0 | 5.0 | 3.0 | 3.5 | 4.0 | 2.5 | 3.0 | 3.5 | 2.5 | 2.8 | 3.0 | 2.0 | 2.5 | 3.0 | 1.5 | 2.0 | 2.3 | 1.5 | 1.7 | 2.0 |
| | 20 | 9 | 4.5 | 6.0 | 7.0 | 5.0 | 6.0 | 7.0 | 4.5 | 6.0 | 6.0 | 4.0 | 5.5 | 6.0 | 3.0 | 4.0 | 5.5 | 2.0 | 3.0 | 4.0 | 2.5 | 3.0 | 4.0 |
| | 15 | 9 | 5.4 | 7.2 | 8.4 | 6.0 | 7.2 | 8.4 | 5.4 | 6.6 | 7.8 | 4.8 | 6.6 | 7.2 | 3.6 | 4.8 | 6.6 | 2.4 | 4.2 | 4.8 | 3.0 | 3.6 | 4.2 |
| 6 | 25 | 22 | 8.2 | 10.5 | 13.0 | 9.0 | 11.3 | 13.0 | 8.2 | 9.8 | 12.0 | 7.5 | 9.7 | 10.5 | 6.0 | 8.3 | 9.7 | 3.8 | 6.0 | 6.8 | 4.5 | 5.3 | 6.7 |
| | 35 | 43 | 10.0 | 12.0 | 16.0 | 11.0 | 12.0 | 15.0 | 10.0 | 12.0 | 14.0 | 9.8 | 12.0 | 14.0 | 7.0 | 9.8 | 12.0 | 5.0 | 7.7 | 9.1 | 5.6 | 6.3 | 9.1 |
| | 15 | 11 | 7.8 | 9.8 | 12.0 | 8.5 | 10.0 | 12.0 | 8.0 | 9.1 | 11.0 | 6.6 | 9.1 | 10.0 | 5.2 | 7.1 | 9.1 | 3.8 | 5.8 | 6.5 | 3.9 | 4.2 | 6.5 |
| 8 | 25 | 30 | 12.0 | 15.0 | 19.0 | 12.0 | 16.0 | 18.0 | 12.0 | 14.0 | 16.0 | 10.0 | 14.0 | 15.0 | 7.8 | 11.0 | 14.0 | 5.2 | 8.5 | 10.0 | 5.8 | 7.1 | 9.1 |
| | 35 | 58 | 14.0 | 18.0 | 22.0 | 14.0 | 19.0 | 20.0 | 14.0 | 16.0 | 19.0 | 12.0 | 16.0 | 18.0 | 9.8 | 13.0 | 16.0 | 6.5 | 10.0 | 15.0 | 7.0 | 8.5 | 12.0 |
| | 15 | 14 | 9.0 | 11.0 | 13.0 | 9.0 | 11.0 | 13.0 | 9.0 | 10.0 | 12.0 | 8.0 | 10.0 | 11.0 | 6.0 | 8.4 | 10.0 | 4.0 | 6.6 | 7.3 | 4.5 | 6.7 | 8.0 |
| 10 | 25 | 38 | 13.0 | 18.0 | 22.0 | 13.0 | 18.0 | 20.0 | 14.0 | 15.0 | 19.0 | 12.0 | 16.0 | 18.0 | 8.8 | 13.0 | 16.0 | 6.2 | 9.6 | 11.0 | 6.5 | 9.5 | 12.0 |
| | 35 | 73 | 15.0 | 20.0 | 24.0 | 16.0 | 21.0 | 23.0 | 15.0 | 18.0 | 22.0 | 14.0 | 18.0 | 20.0 | 11.0 | 14.0 | 18.0 | 7.3 | 11.0 | 13.0 | 8.0 | 12.0 | 16.0 |
| | 15 | 17 | 9.0 | 12.0 | 15.0 | 9.7 | 13.0 | 15.0 | 9.0 | 11.0 | 13.0 | 9.0 | 11.0 | 12.0 | 6.5 | 9.0 | 12.0 | 5.0 | 7.1 | 8.5 | 4.0 | 6.0 | 8.0 |
| 12 | 25 | 45 | 14.0 | 19.0 | 23.0 | 16.0 | 20.0 | 22.0 | 14.0 | 16.0 | 20.0 | 13.0 | 17.0 | 19.0 | 9.3 | 14.0 | 17.0 | 6.6 | 10.0 | 12.0 | 7.0 | 9.0 | 12.0 |
| | 35 | 88 | 17.0 | 23.0 | 29.0 | 19.0 | 25.0 | 27.0 | 16.0 | 20.0 | 25.0 | 16.0 | 22.0 | 23.0 | 12.0 | 17.0 | 21.0 | 9.0 | 13.0 | 14.0 | 8.0 | 12.0 | 16.0 |

## CHAPTER V

## THE AUTOMATIC REFRIGERATING MACHINE

The larger refrigerating machine would necessarily require an operator for part or full time in order to maintain proper operating conditions and to make such adjustments as the machinery requires, but the smaller machines have to be designed in such a way that they will be automatic. This means that they must be capable of starting and stopping under the conditions of temperature in the refrigerated tank or room and that there be a regulator for the water (if water cooling is used) and for the suction pressure, in addition to various other safeguards. Under these conditions the only need for an operating engineer is to prevent clogging of valves, to maintain satisfactory lubrication and to watch against leaks or such troubles as might interfere with the proper functioning of the refrigerant. Development of such automatic machines results in an immediate expansion in distribution, not only to the small retail trader of meats and provisions, the florist and caterer, but to the soda fountain and small ice cream retailer and to household trade. Automatic refrigeration has made the 2, 1 and the fractional tonnage compressor practical.

**The Direct Expansion System.**—The very small compressor is designed so that it may be operated with the least possible attention. Such a compressor is always electric motor driven, the motor being either the induction or the repulsion-induction type as a rule. Automatic operation means that the compressor will not operate continuously, but intermittently with the demands of the load. There would be required in the automatic plant then:

*a.* A temperature control, set for the average temperature of the brine to be cooled, or of the air in the cold storage rooms. This is usually secured by means of a *thermostat* connected electrically with the necessary relays or devices to start or to stop the motor.

*b.* A suction pressure regulator, designed so as to maintain a constant suction pressure and therefore a constant boiling temperature of the refrigerant in the evaporating coils. In design this regulator is practically a pressure reducing valve.

*c.* A water control valve, so as to regulate the condensing water to periods of operation only, unless the condenser is air cooled.

In addition to the required valves already mentioned for the complete automatic machine there must be certain special features in order to safeguard against trouble. These include the excess pressure relief valve, and frequently a liquid valve to prevent the flooding of the evaporator coils during the period of shut down of the compressor. The compressor needs to be automatically lubricated, with special provision against the matter of *oil pumping,* and the shaft packing (such a machine is always of the vertical enclosed type if it is of the usual reciprocating type of design) must be tight against leakage of the refrigerant and yet not subject to much friction. The compressor should be provided with an equalizer connection so that on starting the compressor the pressure on the two sides of the cylinder will be the same, thus permitting the motor to get up to speed before full resistance to operation will take effect and thereby assisting in the decrease of electric motor troubles.

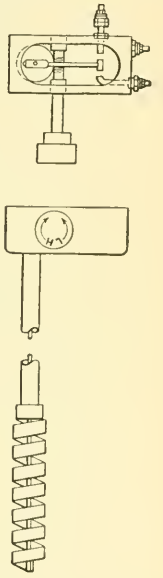


Fig. 102.—Engle-hart Thermostat.

**The Thermostat.**—The thermostat may be of the dissimilar metal type (Figs. 102 and 103) which depends on the variable expansion and contraction of the two metals used with the temperature change of the fluid to which it is exposed or it may be designed to utilize the effect of a volatile liquid exposed to a variable temperature. In the first example, the change of temperature causes a motion of the contact point which either makes contact with the *start* or the *stop* coils of the solenoid control.[1] This thermostat is placed directly in the fluid to be regulated in temperature. There are three different designs:

*a.* Two flat strips intimately joined together are coiled into a helix, one end of which is attached to a central rod and is free to turn (Fig. 102). The twisting moment of the rod is caused by the expansion and contraction of the coil with rise and fall of the temperature.

*b.* A bi-metallic thermo element in the form of a spiral (Fig. 103) forces a blade supported on knife edges so as to make contact at either the right or the left of the vertical, thereby energizing the necessary coils causing the motor to start or to stop.

[1] If the motor is properly designed with the necessary starting torque and the compressor is provided with a pressure equalizer, the compressor will be controlled satisfactorily.

*c.* The sylphon type of thermostat contains a liquid of the proper boiling temperature so as to secure the required differential pressure with a small change of temperature. The bulb (Figs. 104 and 105) may be placed in the cold room or the brine and a flexible tube connection may be made with the bellows. The mercoid control shown in the illustrations can be adjusted to make and break contact with a differential temperature of from 3 to 4 deg. F. The action of the sylphon is to tilt the mercury tube. The tube contains mercury and an inert gas

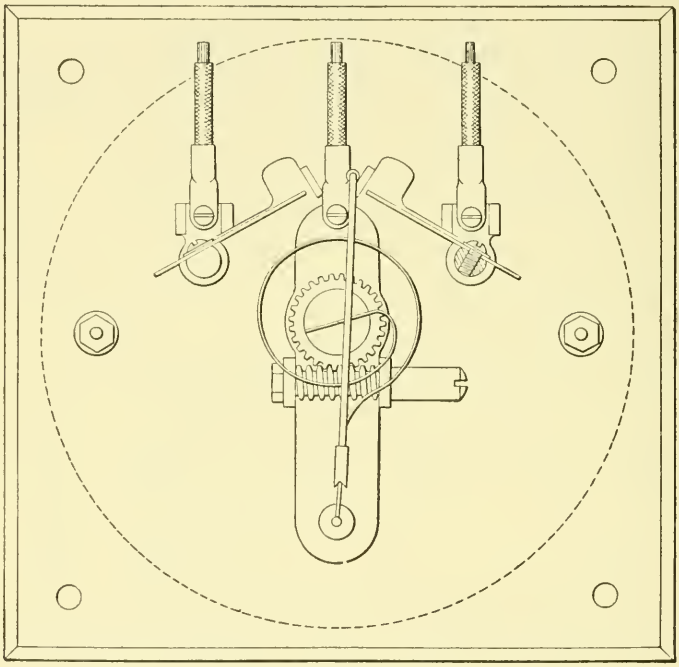


Fig. 103.—Monogram Full-line Voltage Thermostat.

to stifle the electric arc on *breaking*. The normal current is 10 amperes at 110 volts and 5 amperes at 220 volts on either alternating or direct current. This form of thermostat is very successful, as the bellows and the mercury tube have been tested to over one-half a million operations without signs of failure.

**The Automatic Expansion Valve.**—As previously mentioned, the automatic expansion valve is simply a pressure reducing valve. In the usual design it is of the diaphragm type (Figs. 105 to 110), whereby the movement of the diaphragm opens and closes the passage in the valve and permits a flow of the liquid refrigerant from the high-pressure side

to the evaporating coils. One side of the diaphragm is subject to the reduced pressure of the refrigerant, while the other side has the pressure

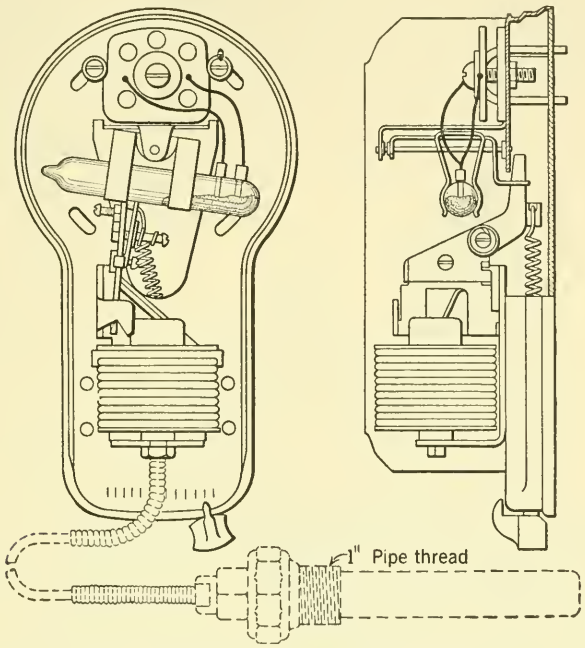



Fig. 104.—Mercoid Type Thermostat.

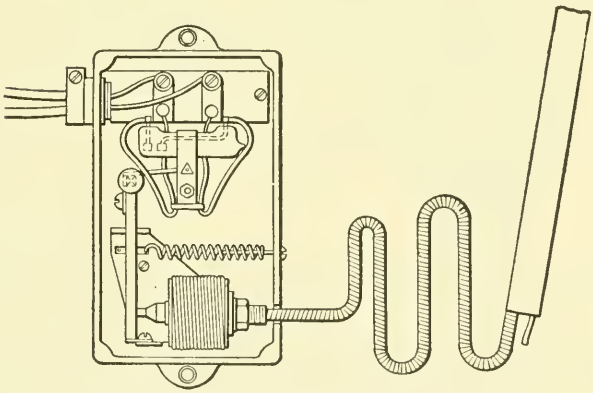


Fig. 105.—Absolute Contactor Thermostat.

exerted by a helical coil spring. This spring has a means of adjustment of the tension making it possible to obtain a control of the reduced (the suction) pressure over the usual operating range.

Such a valve as has just been described of necessity resembles the standard manual control expansion valve. In each there must be the same general design of piston seat and seat disc with provision for minimum wear and with a conical seat so that close adjustment of the flow of the refrigerant may be secured. In one design the piston seat and the seat disc are made of 30 per cent nickel steel, the diaphragm is made of monel metal and the body of the valve of close-grained cast iron. The Cash valve (Fig. 109) and the Peerless valve (Fig. 110) are examples of the construction just described. It is very essential that a scale trap be made an integral part of the valve or that one be placed adjacent to the valve and on the condenser side of it in order to prevent mill scale, sand, etc., from lodging on the valve seat. The Brunswick - Kroeschell valve (Fig. 111) obtains a sliding action which tends to free itself of any scale, and it is besides designed for complete closing as soon as the suction pressure increases above the amount at which the valve is set, thereby tending to prevent the flooding of the evaporating coils with liquid refrigerant during periods of shutdown of the compressor.

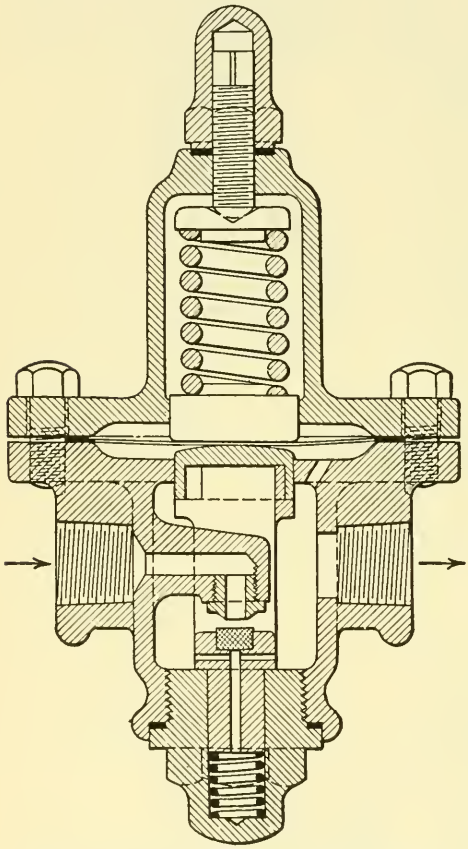


Fig. 106.—The Foster Expansion Valve.

The general criticism of automatic expansion valves is that they are likely to stick on account of the formation of a cold gummy material or ice on the the valve seats which prevents them from functioning properly. The Climax valve (Fig. 112) is entirely different in design from those already described. The diaphragm is replaced by a sylphon subject to the evaporator temperature. Change of temperature causes a breathing of the sylphon which causes a lever (the free end of which is

continually oscillating) to move axially. For certain positions of the lever engagement is made with a pin which causes the valve to open and close with great rapidity while at other positions of the cam the valve will remain closed entirely. It is evident that the suction pressure controls the amount of opening of the valve, and that no sticking of the valve can be possible. There is the objection, however, that the expansion valve must be attached to the compressor.

**Water Control.**—The automatic compressor requires a control on the water, both as regards the amount of the water used and the adjust-

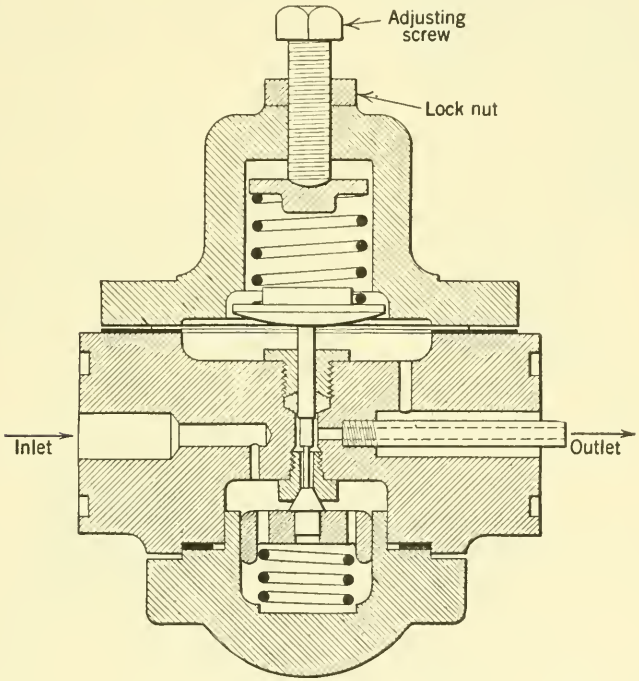



FIG. 107.—Alco Expansion Valve.

ment of the water to the actual periods of operation. There are a number of different designs. Some machines have used direct connection to the shaft of the compressor so that when the compressor is in operation the condensing water will flow, but this form of water valve gives no control of the amount of water being used. Another type (Fig. 114) makes use of a sylphon with the valve placed at the exit from the condenser, so arranged as to have a slight flow even at periods of shutdown. The temperature of the water passing the sylphon affects the amount of the opening through which the water

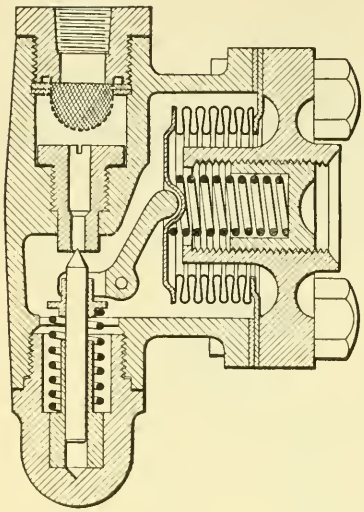


FIG. 108.—American Radiator Expansion Valve.

may pass and thereby permits a greater or lesser amount of water to pass, depending on the initial temperature of the water and the load on the condenser.

The more common type of water regulator valve is shown in Fig. 115 (the Cash valve) and in Fig. 116 (the Monogram valve). In each of these the pressure of the refrigerant is exerted on a diaphragm loaded on the upper part by means of a helical spring which is set for some particular unit pressure, for example, 150 lb. per square inch. When the compressor starts up the condenser pressure increases until it gets to an amount corresponding to that at which the valve was set, whereupon the valve will open and water will begin to flow. The amount of water passing through the valve, and through the condenser, will increase with an increase of pressure until the capacity of the valve is reached. Some adjustment for the individual installation can be obtained by varying the relative position of the spring yoke through the use of the nuts on the standard or by varying the length of the valve stem.

An excess pressure, caused by a failure of the condenser water supply, can be safeguarded against by a pop-relief valve, but such a method is not usual in automatic refrigerating machines, even to discharge the gas into the low-pressure side of the system. A better method of

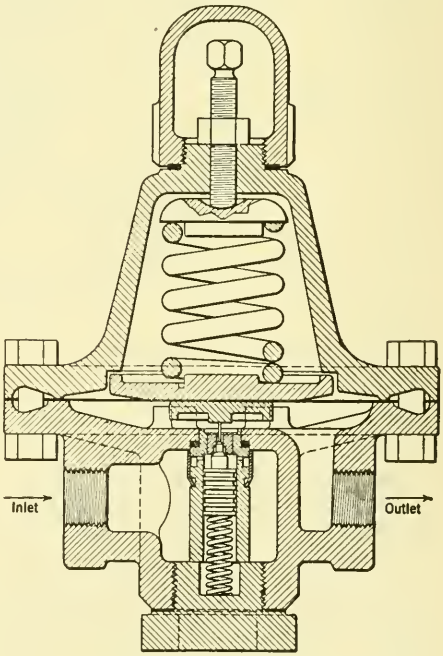



FIG. 109.—Cash Expansion Valve.

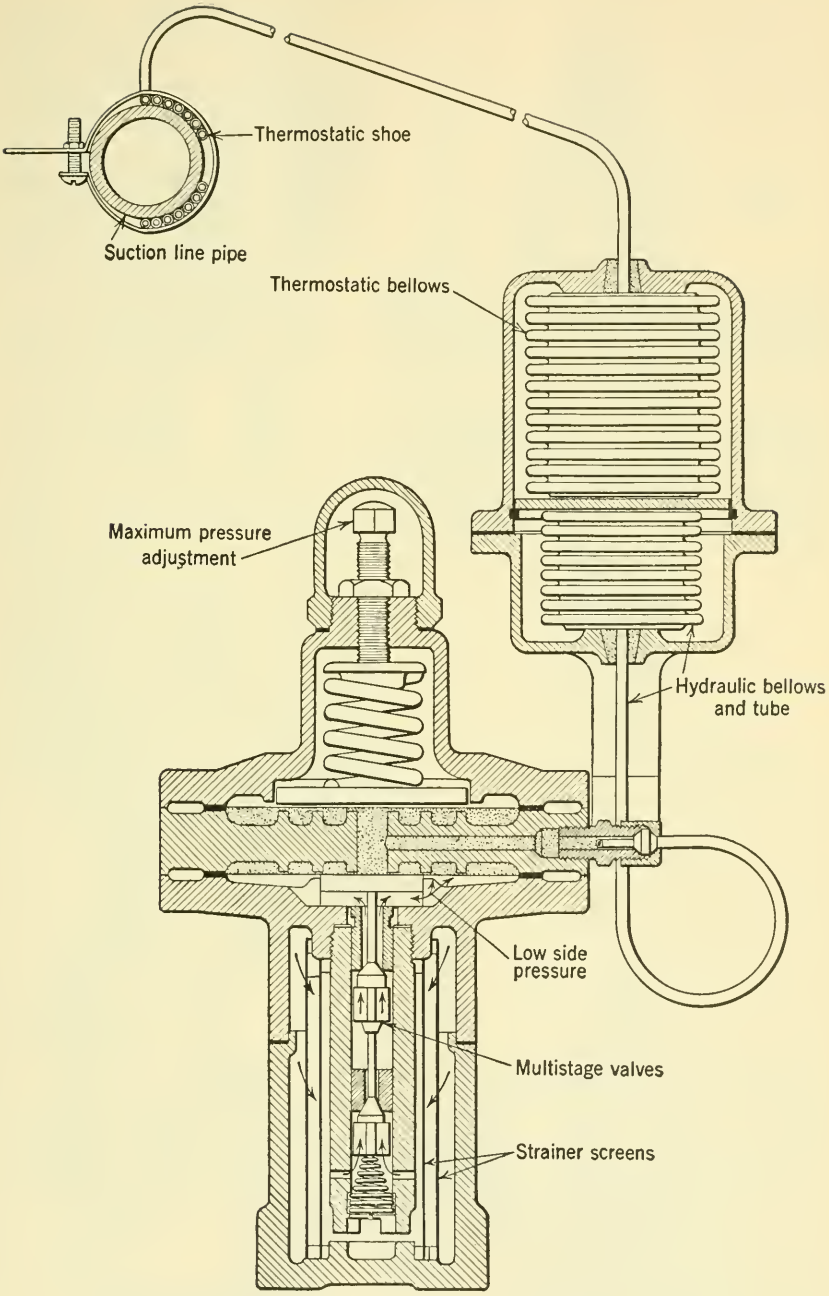



FIG. 110.—Peerless Economizer Valve.

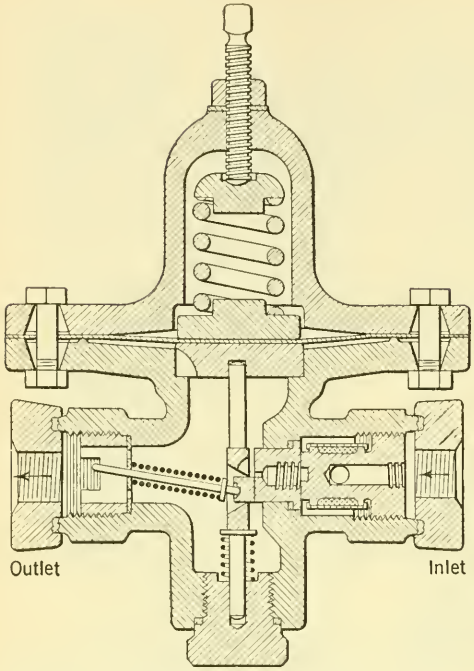



Fig. 111. — Brunswick-Kroeschell Expansion Valve.

Consideration up to the present has been given only to those plants where there is but one set of evaporating coils. The problem is much complicated by having two or more such coils, the loads of which are variable and entirely independent of one another. For such an automatic plant, it is evident that the compressor cannot be shut down until the conditions in all the evaporating coils are satisfied. The thermostatic control to each room, can not be attached to the compressor but only to the control for small machines is to stop the compressor until such time as the pressure decreases to the allowable working amount. In larger machines we may have a condenser pressure circuit breaker, which has to be controlled manually.

Figure 117 shows one type of water pressure failure switch. The water pressure is exerted on one side of the rubber diaphragm, the other side of which has the pressure of a light coil spring. The valve stem motion is communicated by means of a toggle joint to the electric connection, which can be of the knife-edge switch design.

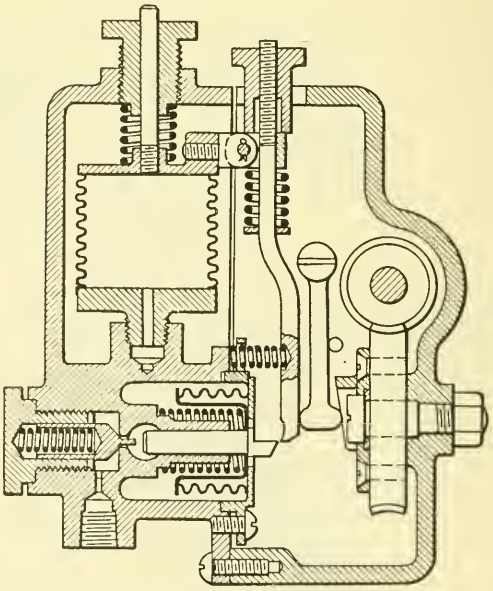


Fig. 112.—The Climax Expansion Valve.

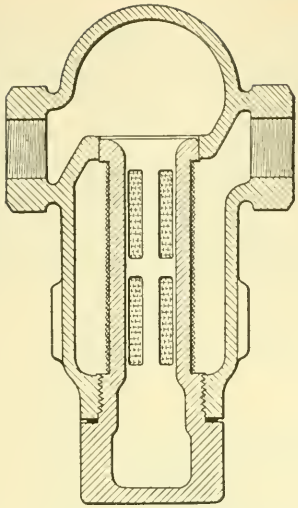


Fig. 113.—Cash Expansion Valve Strainer.

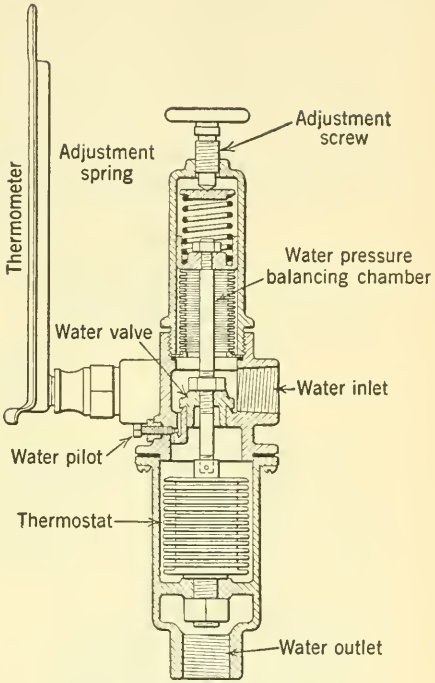
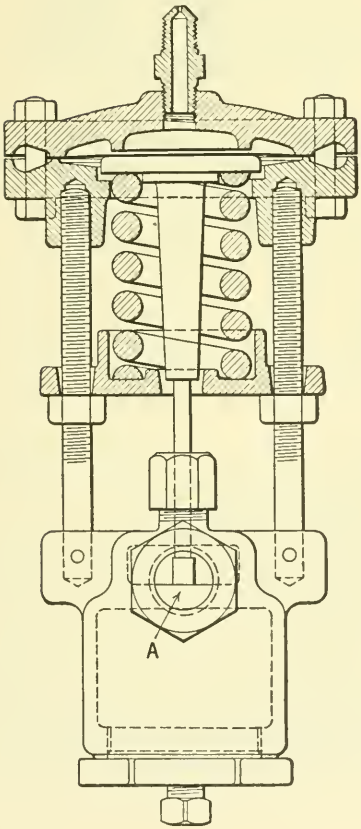



Fig. 114.—Water Regulator.



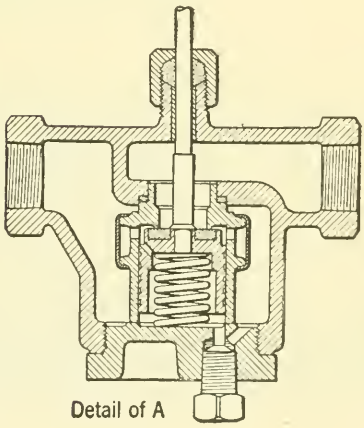




Fig. 115.—Cash Water Regulator.

controls to the individual piping. The compressor operation now will need to be obtained, not from the temperature of the rooms, but from the drop of the suction pressure in consequence of stopping the liquid feed to *all the evaporator coils.*

Figure 118 shows a Monogram type of magnetic stop valve for the regulation of the flow of liquid to a set of evaporator coils (in conjunction with an automatic expansion valve). The thermostat energizes the relay and through the relay the magnet operates the toggle joint.

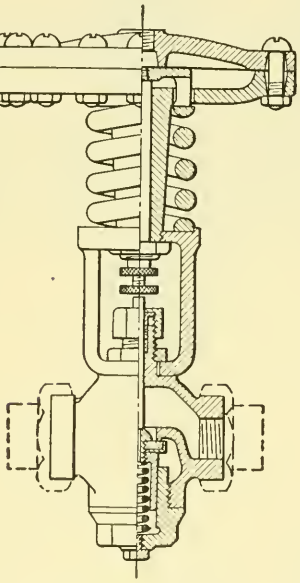


Fig. 116.—Monogram Water Regulator.

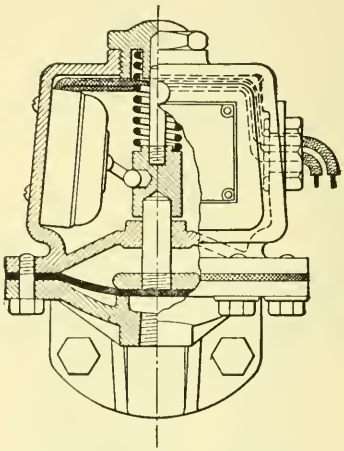


Fig. 117.—Water Pressure Failure Switch.

As soon as the liquid flow is stopped by the closing of the liquid valve the absorption of heat by that particular evaporator coil will stop as quickly as the liquid refrigerant in the coal is vaporized.

Figure 119 shows a form of an ammonia pressure-actuated switch to be used on the suction side in order to control the operation of the compressor where several evaporating coils are used. This valve can be set to operate at some pressure lower than that required during the operation—for example at 5-lb. gage. Such a pressure could not be obtained with any of the evaporator coils in operation if the expansion valves are set at 15 or 20 lb. gage. This suction pressure could only be obtained when all the coils are shut off from the liquid refrigerant.

at which time the compressor will function only to pump down the low pressure system. Figure 120 shows a multiple thermostatic control for

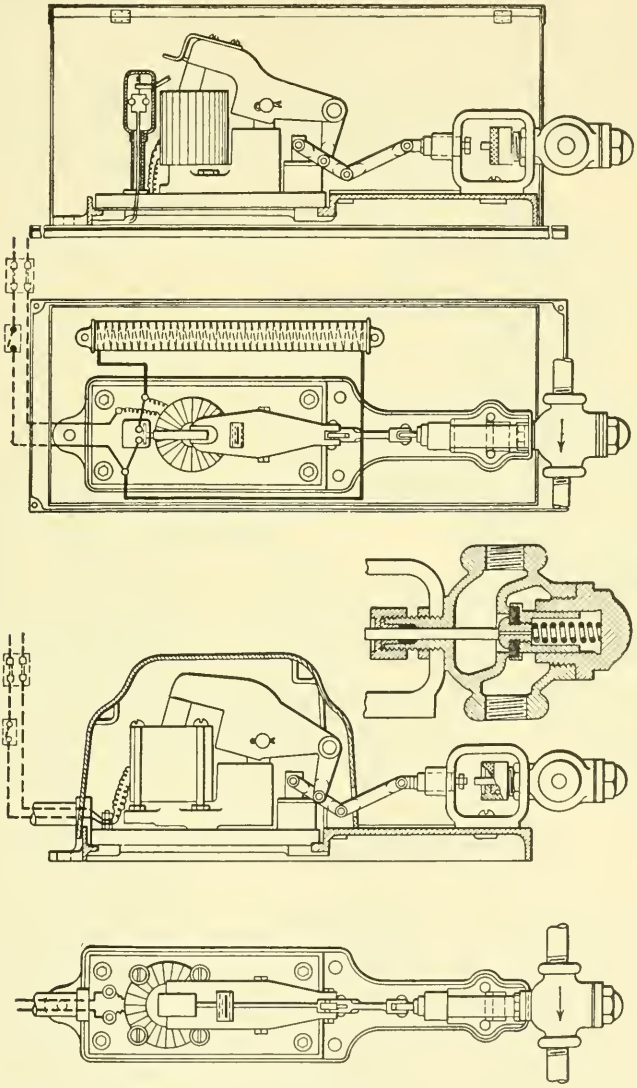


Fig. 118.—Magnetic Stop Valve.

two or more evaporating coils using the controls already described and illustrated.

Somewhat different from what has just been described is the Hilger No-freeze (Fig. 123) back valve. In essentials the liquid, after leaving

the automatic expansion valve, passes through the upper part of the no-freeze back valve. The suction gas on its return has to pass through the lower part of the valve which is subject to the suction gas temperature in such a way[2] as to operate a valve in the liquid line, even to the point of complete closure.

**Dual Thermostats.**—In laying the piping for automatic plants some very interesting arrangements are made at times. An example of such a design might be the arrangement prepared for the florist who has a

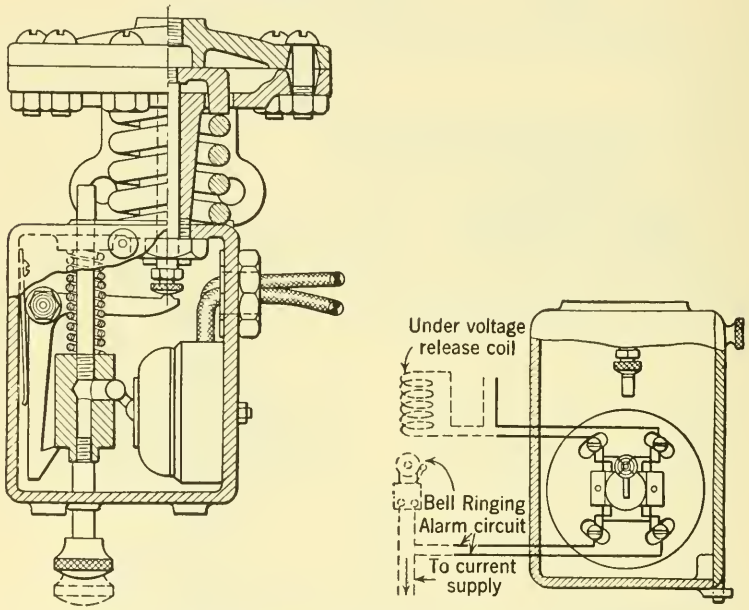



Fig. 119.—Ammonia Pressure-actuated Switch for Maximum Pressure.

show case and a cold storage room, with the piping connected in series, with a thermostat in each. The liquid enters the coils in the show case, and on leaving these coils passes to the piping in the cold storage room. The show case has a thermostat set at 45 deg. F., and the cold storage room has one set at 35 or 36 deg. F. The second thermostat acts only to protect the flowers in the cold storage room from freezing should an extreme condition develop where the cold storage room is cooled to a dangerous temperature before the first thermostat has a chance to act.

[2] The decreased temperature of the ammonia gas contained in the center of the valve exerts a decreased pressure on the diaphragm and the diaphragm is pressed downward by the fluid pressure on the upper part.

**Brine Circulation.**—Where brine is circulated the automatic features are relatively simple. Such a plant would have brine storage, and in the simple arrangement the thermostat would start or stop the compressor for a certain brine temperature range. Such a control is obtained easily through the mercoid or other form of thermostat switch. Where the brine system consists of a plurality of coils, each of which requires a control, there will need to be a thermostatically controlled valve on each coil equipped with solenoids in order to give the proper opening and closing required (Fig. 124).

**The Small Refrigerating Machine.**—The small refrigerating machine is essentially automatic. It is inconceivable that the fractional tonnage machine, used now so extensively for ice cream cabinets, small boxes, and for the household refrigerators, could be operated in any other manner. To a lesser extent, perhaps, the same thing is true of the 1, 2, 4, 5, and even the 10-ton machine as the demand is increasing for a machine in these sizes which does not require constant attention from an expert operator. It is appropriate, therefore, to

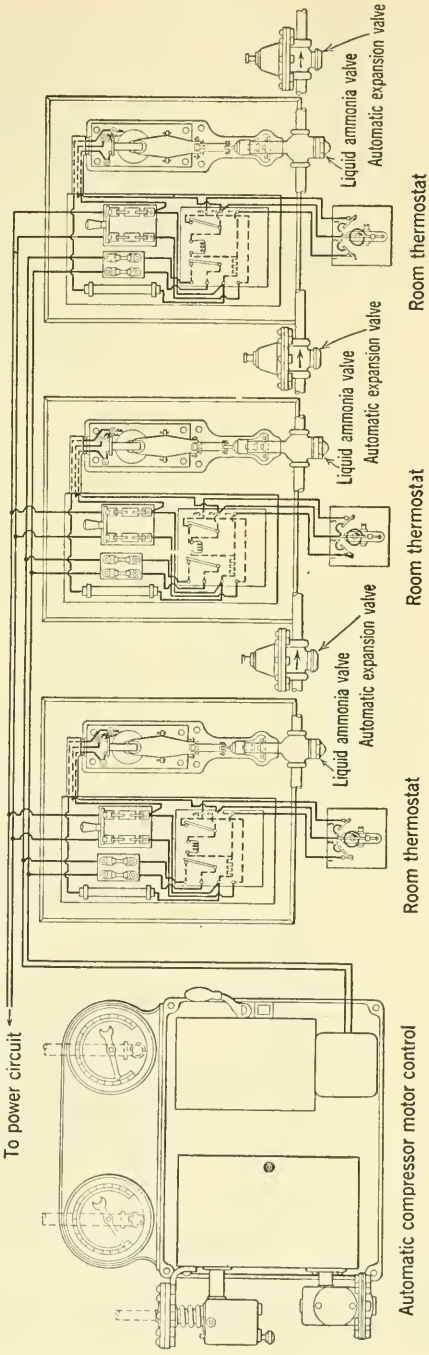



FIG. 120.—Method of Automatic Control.

take up these small machines an an example of the automatic features just described. In all probability no other type of mechanical refrig-

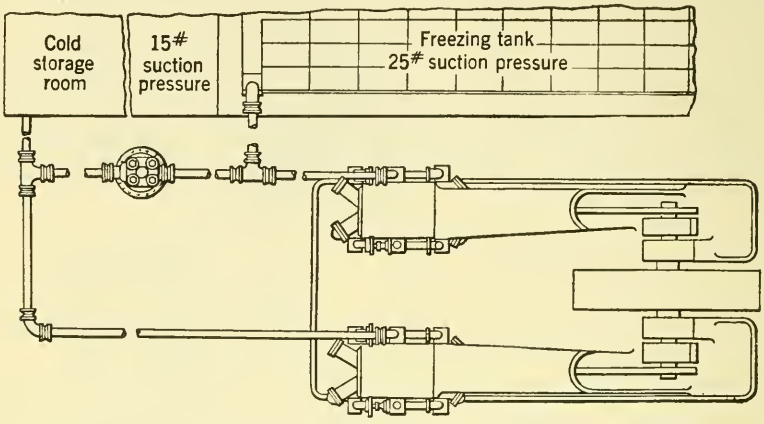



Fig. 121*a*.

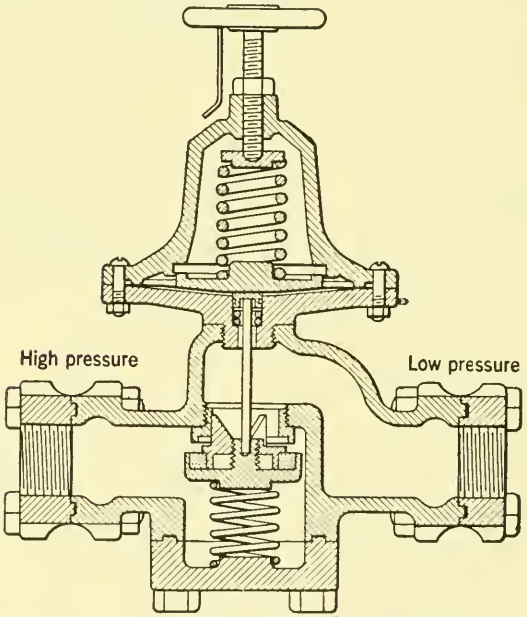



Fig. 121*b*.

Fig. 121.—Ammonia Pressure Regulator.

eration is attracting the interest that the small machine does, and its popularity derives from the following reasons:

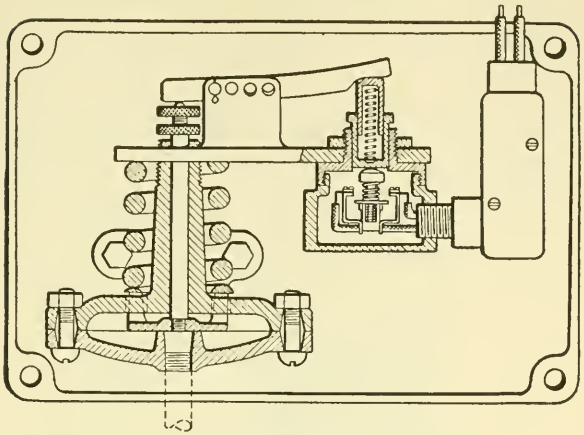


FIG. 122.—Pressure-actuated Switch.

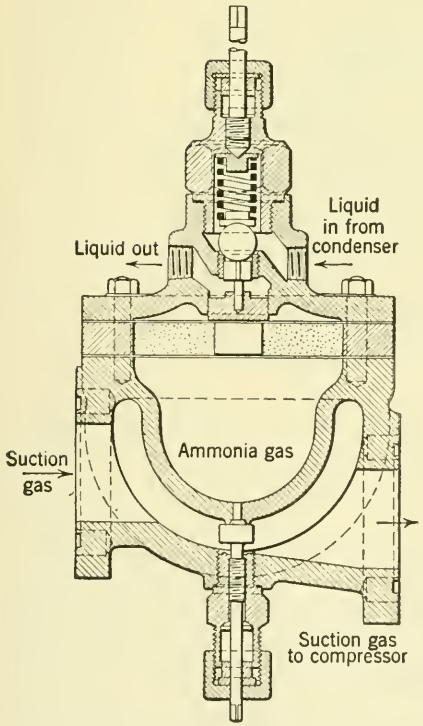



FIG. 123.—Hilger Non-Freeze Back Valve.

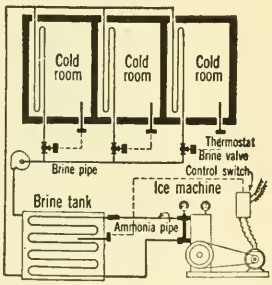



FIG. 124.—Thermostatic Control of Brine.

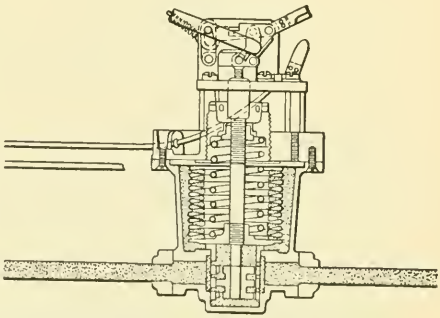


FIG. 125.—Pressure Control on Household Compressor.

Ice for small cold storage boxes or refrigerators is frequently found to be dirty and unreliable. At times the ice delivery service has not been of the best, and some localities have not been able to secure ice at all. Besides this, the temperature in the boxes—for storage without the use of salt on the ice—has been found to be unsatisfactory in many cases. The refrigerators may be designed poorly or be in poor condition, with the result[3] that temperatures are not lower than 45 to 55 deg. F.

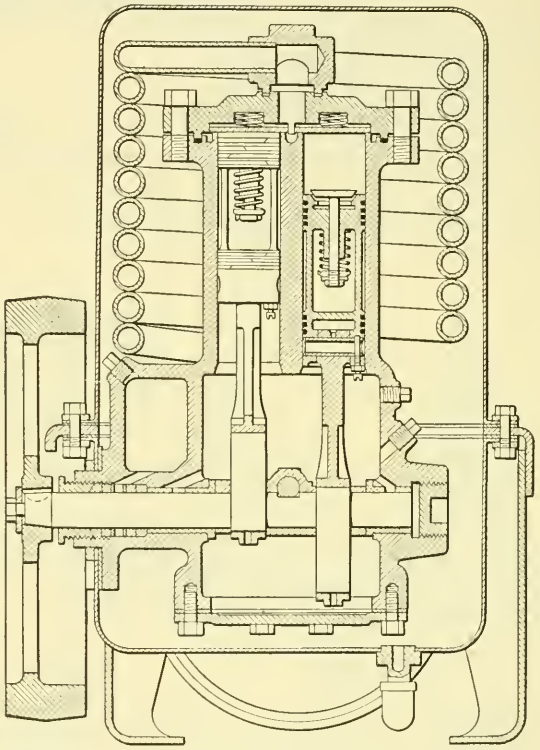


Fig. 126.—One-half to One-quarter-ton Compressor.

In general, it can be said that temperatures in such a box cannot be obtained lower than 35 and are more often greater than 40 deg. F

The mechanically cooled refrigerator gives practically any temperature desired, controlled thermostatically within a slight range. At times it gives a drier box, although good circulation will do the same

[3] The Transactions N. E. L. A., 1923, give results of tests by Prof. A. M. Greene Jr., in which it was found that only 4 per cent of 250 ice-cooled refrigerators which were tested operated with a temperature less than 45 deg. F.

for the ice-cooled refrigerator. The variations in the temperature at different times of the day are very slight as compared with ice cooling, for the latter requires a full supply of ice in the ice compartment at all times in order to get good results.

As regards costs of operation, the real household refrigerating machine cannot compete with ice, if ice costs are those usually found in the larger cities. The operation cost, which must include the cost of power, water (if used), interest and depreciation on the mechanical equipment, repairs, etc., is usually an appreciable amount, and frequently will be found to be about twice the cost of the operation of the

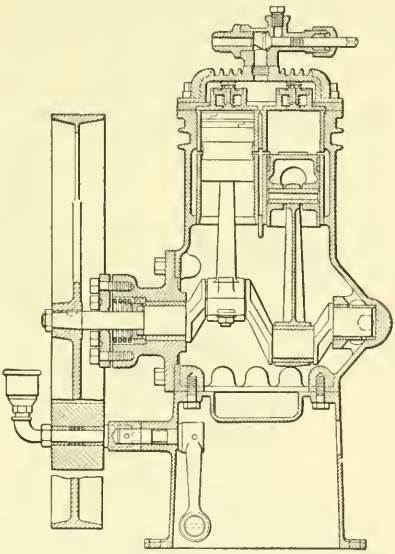


Fig. 127.—Kelvinator Compressor.

ice-cooled refrigerator, especially in those localities where ice does not have to be used for the entire 12 months. There are a number of automatic features in the household refrigerating machine; the thermostat, the excess pressure cut-out, and the suction pressure control. If the condenser is water cooled, the machine should have some means of regulation of this water. The companies selling service during the first year of operation consider that they are lucky if an average of only 2.5 calls per year per machine are required. This service is required to adjust the automatic features as well as to attend to the more serious troubles due to the loss of the refrigerant, loss of oil, or to failure of the expansion valve to operate satisfactorily. Most concerns selling these small machines provide free service for the first year of

operation, and after the first year render service at about $1 per month. Without question these small compressors are becoming more perfect every year, and their popularity is increasing rapidly in spite of the undoubted extra cost. This statement does not apply to the $\frac{1}{6}$-ton machines which are used to a large extent for ice cream cabinets where temperatures of 10 deg. F. or lower are required.

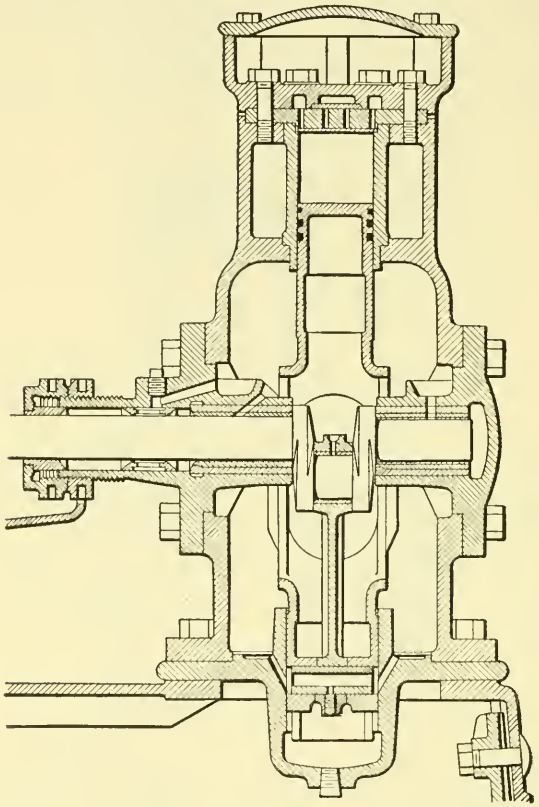


Fig. 128.—Universal Compressor.

**The Refrigerant and the Operating Conditions.**—The usual refrigerants are ammonia, sulphur dioxide, ethyl and methyl chloride, butane, isolutane, propane, etc. Carbon dioxide is not used as a refrigerant for household compressors because of its excessive pressure. By means of the automatic suction pressure control and the use of a suitable thermostat, the temperature carried in the box can be set at will, and is usually set at 35 to 40 deg. F., with a variation of about 2 degrees on either side of the desired temperature. With the exception of the absorption

machine, the household refrigerating machine is always electrically driven, using a $\frac{1}{4}$-hp. motor, and the power consumption is about 1.5 kw.-hr. per 24 hours. The actual running time is from 5 to 9 hours, and the compressor starts and stops approximately 6 to 10 times per 24 hours. At present there are some air cooled, up to about

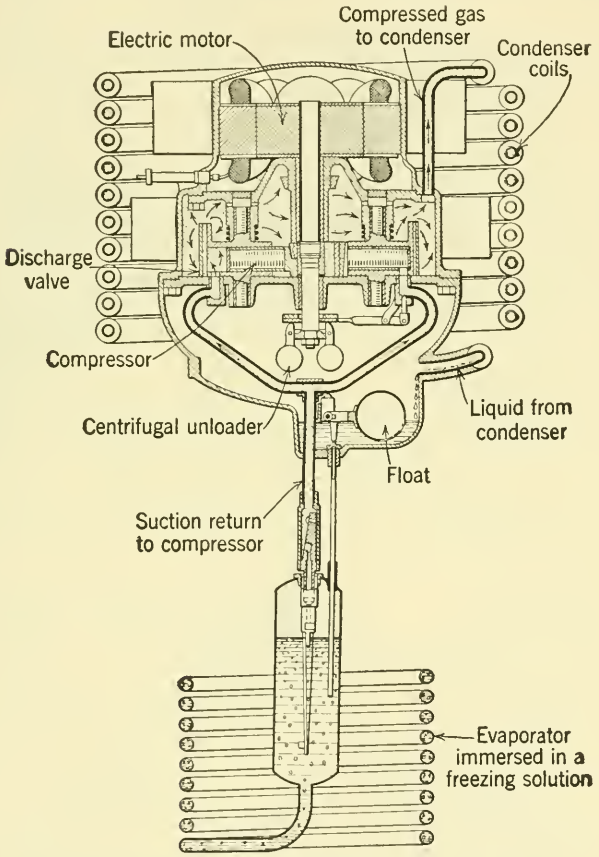



Fig. 129.—The General Electric Compressor.

$\frac{1}{2}$-ton capacity, and some water-cooled condensers. The air-cooled condenser has a higher temperature of liquefaction of the refrigerant with a temperature of liquefaction of the working medium some 10 to 20 deg. higher than the room air temperature, and therefore more power is required of the compressor than with a similar case using tap water over the condenser. In addition, the air-cooled condenser requires a fan to drive the air through the condenser, and this is an added load to

the motor, but usually not exceeding 0.02 hp. on the $\frac{1}{4}$-ton size. In the water-cooled condenser the water consumed varies from 10 to 15 cu. ft.

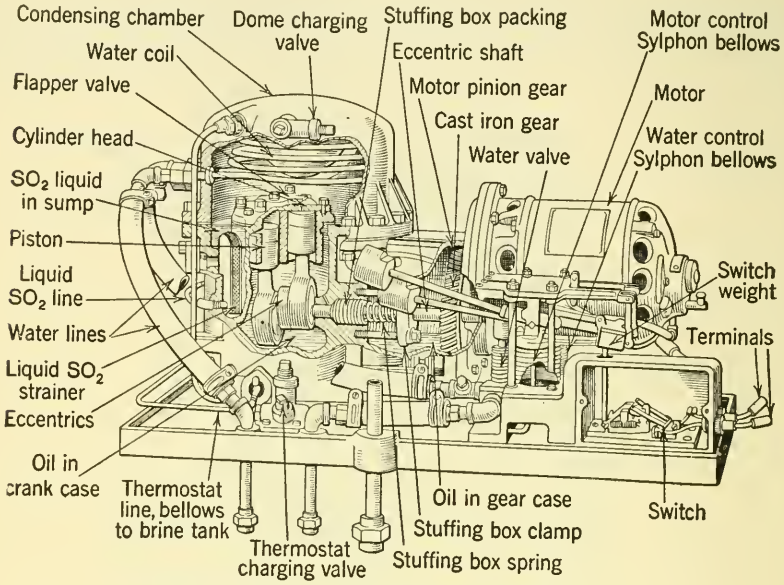



Fig. 130.—The Frigidaire Compressor.

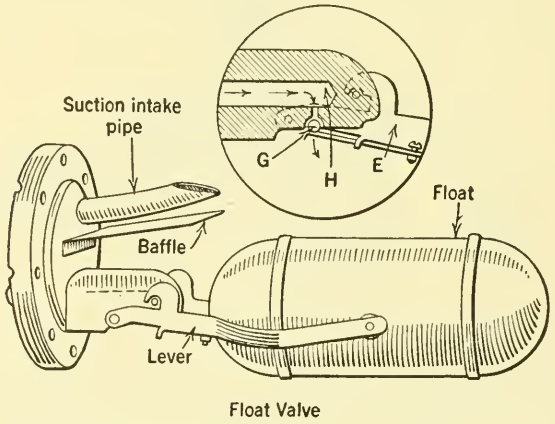



Fig. 131.—The Float Valve for Household Compressors.

per day. However, the air-cooled condenser has a great advantage in being simpler in design, is not dependent on the continuity of the water supply, and appears to be more and more popular.

The expansion valve may be of two principal designs—the diaphragm type (Fig. 125) and the float type (Fig. 131). In the household machine the low-pressure side is relatively small and the charge of the refrigerant is relatively quite nominal. This is the reason why the float type of valve can be used and in the sulphur dioxide compressor this kind of expansion valve works out best as regards the lubrication of the compressor. The diaphragm type of compressor can be set for any pressure, and this is usually at such an amount as will give a brine temperature of from 10 to 20 deg. F. The main objection to the diaphragm type of

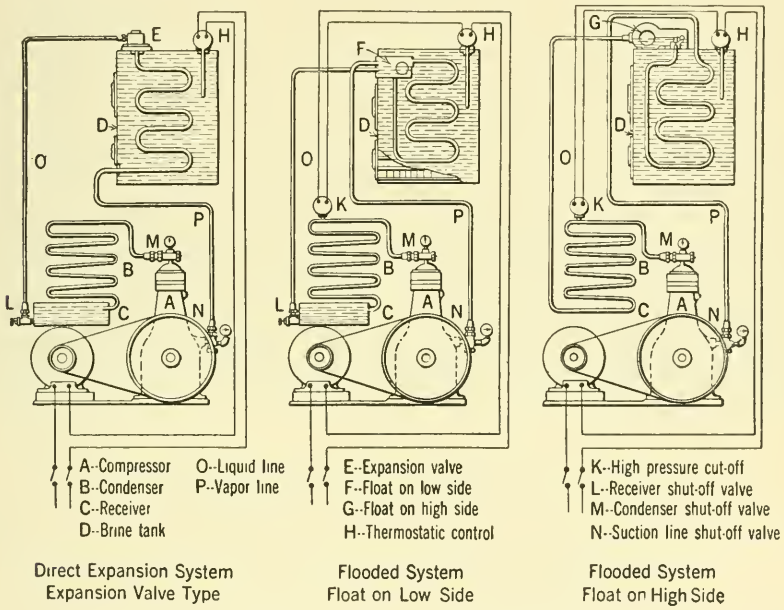



Fig. 132.—Typical Piping Arrangements—Household Compressors.

valve is that when the compressor is stopped the valve frequently will permit the flooding of the evaporating coils, creating a condition that will require some time to pump down after the compressor is started again, thus resembling the action of the float type of valve.

The thermostat is usually designed with the sylphon containing a volatile liquid-like sulphur dioxide. As the temperature rises this liquid exerts a higher pressure and the the design is such that the increment increase of the pressure will operate the sylphon which operates the control switch. In this manner the motor may be started or stopped. If the condenser is water cooled, this water may be controlled in a number of ways. A sylphon may be attached to the water outlet, in

such a manner that the temperature of the exit water will vary the amount of the water flowing to suit the conditions of the plant, or the water control may be attached directly to the compressor. Still another method is to use a diaphragm type of valve, operated by the condenser pressure. Figures 126 to 130 show typical designs of household compressors, several of which are no longer being built. Figure 132 gives three types of arrangements in common use.

The household compressor or absorption machine has to be designed so as to be very self-contained. In the compressor the design must be such that the oil will not be lost from the system as this would produce a loss of lubrication; nor must there be any loss of the refrigerant to the atmosphere, resulting in a failure of the machine to function, and, in the case of the sulphur dioxide machine, in the corrosion of that metal near where the leak takes place. The automatic controls need to be so well designed that they will function for a period of time without adjustment or repair. Where gas only is available the absorption machine may work in well. This is of the intermittent type, wherein the heating of the generator by the gas flame (or electric resistance) boils out the ammonia gas which is condensed and stored in the liquid receiver. After the proper time-interval the flame is shut off and water is passed through the generator, which is now functioning as an absorber, and the liquid after boiling in the expansion coils returns to the generator again in order to re-enter solution.

## CHAPTER VI

## HEAT TRANSFER

Refrigeration, like other branches of heat engineering, is vitally interested in heat transfer, both from the viewpoint of heat leakage through the building material and from the consideration of the flow of heat through the various parts of the refrigerating cycle. At times the refrigerating load is entirely one of heat leakage, and so the insulation against this leakage is one of the primary subjects for careful thought. In the refrigerating cycle heat transfer is all important in the condenser, brine cooler, or other evaporating surfaces of the low-pressure side, and in the generator, rectifier, exchanger, absorber, etc., of the absorption machine. The problem in many cases is also complicated by the variation in the velocity of the fluid in contact with the surface and by the insulating effect of the air, oil, etc., to be found inside the pipes in nearly every case. Some of these factors will be taken up briefly.

**Theory of Heat Transmission.**—In general, heat transfer may take place in three different ways: by radiation, by conduction, and by convection. In refrigeration, however, the temperature range is small, and it is seldom that the effect of radiation is appreciable.

**Radiation.**—Every surface has a definite emitting power and a definite absorbing power for radiant heat. This radiant heat is transmitted from one body to another by means of ether waves, so that the presence of air (or the lack of it) does not affect radiation materially. The amount of heat transmitted varies as the *fourth* power of the absolute temperature for black bodies. This relation has been shown by Stefan and Boltzmann, expressed in English units, by

$$Q_{\text{radiation}} = RA(T^4{}_2 - T^4{}_1)$$

where

$T_2$ and $T_1$ are the surface temperatures in degrees absolute of the bodies in question,

$A$ is the area in square feet,

$R$ is a constant.

In refrigeration, as radiation by itself plays such a small part, it can be neglected in most cases. Sometimes, however, in reading temperatures, the thermometer may give incorrect readings when exposed to direct radiation from hotter or colder bodies because of this factor.

**Convection.**—Convection losses have to do with a fluid in motion past the surface in question. It is worthy of remembrance, however, that a layer of the fluid in contact with a solid is always at rest because of adhesion, and a finite thickness of the fluid adjacent to the solid—due to friction and the viscosity of the fluid—is always either at rest or is moving too slowly to have any appreciable effect on the resistance to the passage of heat. If the fluid happens to be oil, it appears that the surface film, which is at rest, has the same resistance to the flow of heat as cork-board of the same thickness. The thickness of this film is dependent on the viscosity and the density of the fluid and the velocity of the current next to the stagnant layer. It varies, with air, from nearly 0 to $\frac{1}{2}$ in. Obviously, the value of this skin resistance is not of appreciable importance in surfaces using cork or other heat insulation, but for glass it is a very important factor, although even glass is not used in mechanical refrigeration except in multiple thicknesses and in very special cases.

Peclet's laws for convection, expressed in English units, are given by

$$Q_{\text{convection}} = NV^{-n}A(t_2 - t_1)$$

where

$N$ is a coefficient of convection and a variable,

$V^{-n}$ is some power of the velocity in feet per second across the section,

$(t_2 - t_1)$ is the difference in temperature between the surface and the surrounding air.

**Conduction.**—Conduction is the means whereby heat is conveyed through molecular vibration, under which conception there is established a temperature gradient in the material. It is well established experimentally that if the material is homogeneous, and nominal temperatures are being employed that the heat flow becomes:

$$Q_{\text{conduction in B.t.u. per hour}} = Ak(t_2 - t_1)$$

Where

$A$ is the area in square feet,

$k$ is the coefficient of heat transfer in B.t.u. per square foot per degree difference per hour, and

$(t_2 - t_1)$ is the temperature difference on the two sides of the heat transfer surface in deg. F.

The effect of radiation and convection can be approximated by the method used by Willard and Lichty[1] by means of the following:

If the temperature differences are small, as they are in refrigerating practice, the value of the radiation and convection losses per square foot of surface per hour becomes:

$$Q_{\text{radiation}} + Q_{\text{convection}} = R[(t + 460) - (t_1 + 460)] + NV^{-n}(t - t_1)$$

and for walls small in height, and in still air this becomes

$$Q_{\text{radiation}} + Q_{\text{convection}} = k_1(t - t_1),$$

but where an outside surface is exposed to nominal air velocity this becomes:

$$Q_{\text{radiation}} + Q_{\text{convection}} = k_2(t_2 - t_0)$$

and

$$Q_{\text{conduction}} = \frac{C}{T}(t_1 - t_2)$$

where

$T$ = the thickness in inches, and

$C$ = is the conductivity of the material.

As $Q$ at any part of the wall must be equal to that of any other part of the section,

$$Q = k_1(t - t_1) = \frac{C}{T}(t_1 - t_2) = k_2(t_2 - t_0) = U(t - t_0).$$

Therefore,

$$\frac{1}{k_1} + \frac{T}{C} + \frac{1}{k_2} = \frac{t - t_1}{Q} + \frac{t_1 - t_2}{Q} + \frac{t_2 - t_0}{Q} = \frac{t - t_0}{Q} = \frac{1}{U}$$

and

$$U = \frac{1}{\frac{1}{k_1} + \frac{1}{k_2} + \frac{T}{C}} + \text{ etc.}$$

It is clear from the foregoing that if a compound wall had been considered in the formula it would have become

$$U = \frac{1}{\frac{1}{k_1} + \frac{1}{k_2} + \frac{T_1}{C_1} + \frac{T_2}{C_2} + \frac{T_3}{C_3}} \text{ etc.}$$

This equation is the one generally used in the computation of the heat leakage in buildings, cold storage rooms and boxes, but in order to use this formula the values for $k_1$ and $k_2$ must be known as well as the

[1] Willard and Lichty, Bulletin No. 102 of the Engineering Experiment Station, University of Illinois.

average values of $C$ for building material and insulation. The values for $k_1$ are given in Table 46, and it was recommended by Willard and Lichty that the values for $k_2$ be taken as three times those for $k_1$ for a nominal velocity of the wind of not more than 15 miles per hour.

Values[2] of $C$ for insulation are given in Table 43 and for building material are given in Table 44. For convenience Table 45 is also included, calculated for 1.0 degree difference in temperature, and for the heat transfer per square foot per hour for 20, 40, 60, 80 and 100 degrees difference in temperature on the two sides of the built-up material. In these calculations the values of $k_1$ and $k_2$ are included,

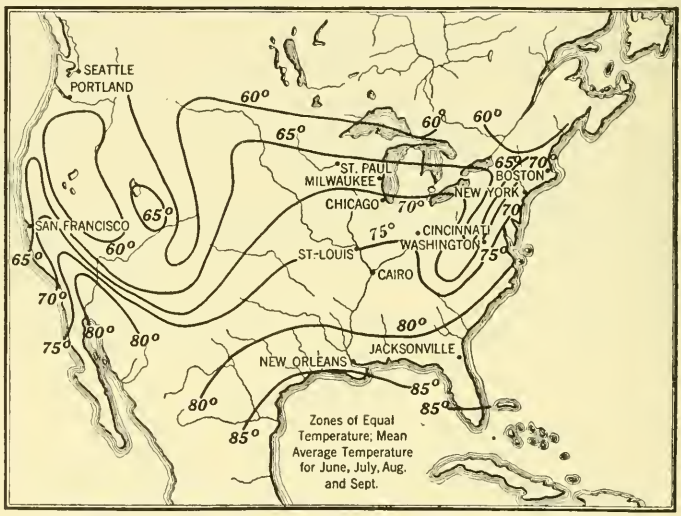



Fig. 133.—Zones of Average Temperature for July, August and September.

although it is very doubtful whether such refinements are justified, especially as the personal equation of the workmen requires that a factor of about 25 per cent be added to the heat transfer as calculated. As a rule the skin effect increases the value of the denominator by 0.77 for each *inside*, and by 0.25 for each *outside* surface. In order to get an idea of the average temperature of the air throughout the United States, average isothermals for the three summer months are given in Fig. 133. In cold storage work it is not the maximum air temperature that should be taken for the calculation of the heat leakage, but the maximum average sustained temperature for three or more days.

[2] A very complete review of insulation values is given in the Report of the Insulation Committee, American Society of Refrigerating Engineers, 1924.

TABLE 43

THERMAL CONDUCTIVITIES OF VARIOUS INSULATING MATERIALS—U. S. BUREAU OF STANDARDS, DEPARTMENT OF COMMERCE

| Material | Conductivity, B.t.u. per Square Foot per Degree Difference of Temperature per Hour | Density | Nature of Material |
|---|---|---|---|
| Air | 0.167 | ..... | Horizontal layer heated from above. |
| Colorax | 0.221 | 0.064 | Fluffy, finely-divided mineral material. |
| Hair felt | 0.246 | 0.27 | Hairfelt confined between layers building paper. |
| Keystone hair | 0.271 | 0.30 | |
| Pure wool | 0.242 | 0.107 | Firmly packed. |
| Pure wool | 0.242 | 0.102 | Firmly packed |
| Pure wool | 0.262 | 0.061 | Loosely packed. |
| Pure wool | 0.292 | 0.039 | Very loosely packed. |
| Cotton wool | 0.292 | 0.100 | Firmly packed. |
| Insulite | 0.296 | 0.19 | Pressed wood pulp, rigid, fairly strong. |
| Corkboard (pure) | 0.317 | 0.18 | |
| Granulated cork | 0.34 | 0.160 | |
| Eelgrass | 0.321 | 0.25 | Inclosed in burlap. |
| Flaxlinum | 0.329 | 0.18 | Vegetable fibres, firm and flexible. |
| Fibrofelt | 0.329 | 0.18 | Vegetable fibres, firm and flexible. |
| Balsa wood | 0.346 | 0.12 | Very light and soft. |
| Rock cork | 0.346 | 0.33 | Pressed rock wool with binder, rigid. |
| Waterproof lith | 0.409 | 0.27 | Rock wool, vegetable binder, not flexed. |
| Pulp board | 0.433 | ..... | Stiff pasteboard. |
| Air cell, ½ in | 0.446 | 0.14 | Corrugated asbestos paper enclosing air spaces. |
| Air cell, 1 in | 0.479 | 0.14 | |
| Asbestos paper | 0.491 | 0.50 | Fairly firm but easily broken. |
| Infusorial earth | 0.579 | 0.69 | (Block). |
| Fire felt (sheets) | 0.596 | 0.42 | Asbestos sheets coated with cement, rigid. |
| Fire felt, roll | 0.637 | 0.68 | Soft, flexible asbestos. |
| 3-ply Regal roofing | 0.696 | 0.88 | Flexible tar roofing. |
| Asbestos mill board | 0.847 | 0.97 | Pressed asbestos, fairly firm, easily broken. |
| Cypress | 0.657 | 0.46 | |
| White pine | 0.792 | 0.50 | |
| Virginia pine | 0.957 | 0.55 | |
| Oak | 1.000 | 0.61 | |
| Hard maple | 1.125 | 0.71 | |
| Asbestos wood | 2.710 | 1.97 | Asbestos and cement. |

The insulation of pipes is given by Heilman [3] in Table 47 and the equation at the bottom of this table. However, the problem is slightly different from that of flat plates (as walls, floors, etc.), and it is usual to take the values of different insulations per one foot of length of nominal pipe size as found by experiment. The thicknesses used are those for drinking water, standard brine and extra thick brine. Values for heat transfer for these thicknesses are given in Tables 48 and 49.

TABLE 44

COEFFICIENTS BASED ON HEAT TRANSMISSION TESTS [B.t.u. per Hour]

(These values are based on the tests run under most satisfactory conditions)

| Material | $C$ per 1 In. Thickness per Square Foot per 1 Deg. F. | $K$ Still Air per 1 Deg. F. |
|---|---|---|
| Brick wall (mortar bond and dry conditions) | 4.00 (5.00) | 1.40 |
| Concrete, 1–2–4 mixture | 8.30 | 1.30 |
| Wood (fir, one surface finished) | 1.00 | 1.40 |
| Corkboard | 0.32 | 1.25 |
| Magnesia board | 0.50 | 1.45 |
| Glass (actual glass 91.4 per cent of total area) | 2.06‡ | 1.5 to 2.00 |
| 2-in. tile, ½-in. plaster on both surfaces | 1.00§ | 1.10 |
| 4-in. tile, ½-in. plaster on both surfaces | 0.60§ | 1.10 |
| 6-in. tile, ½-in. plaster on both surfaces | 0.47§ | 1.10 |
| 2-in. tile, plastered as above and roofing covered | 0.84 | 1.25 |
| Asbestos board | 0.50 | 1.60 |
| Sheet asbestos | 0.30 | 1.40 |
| Double glass, ½-in. air space (glass 69.3 per cent of total area) | 1.50‡§ | 2.00 |
| Roofing* | 5.30§ | 1.25 |
| Air space† | 1.00–1.70§ | |
| Mortar | 8.0 | 0.93 |
| Shavings | 0.42 | |

* Calculated from values of $C$ for 2-in. tile with and without roofing. $\left(\frac{1}{C} = \frac{1}{C_1} + \frac{1}{C_2}\right)$

† See "Air Spaces."

‡ Per square foot of actual glass set in wood frame but based on total heat transmitted.

§ For thickness and construction stated, not per 1 in. of thickness.

**Kinds of Insulations.**—Referring to Table 43, it will be seen that there are a large number of substances that are suitable for low-temperature insulation, but a choice of an insulating material should not be made without all pertinent factors entering into the consideration. For example, the value of $C$ for ½-in. air space is given as 10.7 B.t.u., a

[3] R. H. Heilman, Mechanical Engineering, 1924.

1-in. air space is 11.5 and a 1-in. horizontal layer is given as 4 B.t.u. per hour. It is understood that air is the best insulator known, provided convection can be prevented, and it can be prevented best by the use of a material with a large number of air cells of small size separated from one another as is to be found in cork, hair felt, wood and vegetable material of various sorts, as flaxlinum, fibrofelt and other compressed vegetable fibre.

Some of these materials absorb moisture and need to be protected by means of waterproof paper or they must be made waterproof in the process of manufacture. This statement refers to hair felt, vegetable fibres and woods and to the mineral insulations like rock cork, waterproof lith, asbestos and others of similar nature. Waterproof lith is a mixture of flax fibres, 40 per cent by volume, and limestone rock made waterproof by means of petroleum vapor during manufacture. Sawdust has been used and shavings are still used in the cheaper constructions, in which case the material should be dried and protected from the absorption of moisture by means of waterproof building paper. Granulated cork can be used to advantage in special cases, but as a rule the preferred type of insulation is in a form of board of convenient size for erection on the wall, floors and ceilings. Waterproof lith is made in boards 18 in. × 48 in., and corkboard is usually 12 in. × 36 in. Thicknesses vary, but cork can be obtained from $\frac{1}{4}$ to 1 in. by eighths, and in $1\frac{1}{2}$, 2, 3, 4, and 6 in.

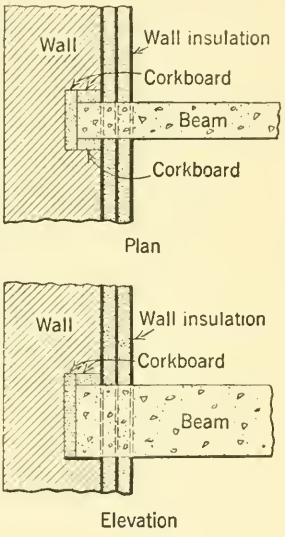
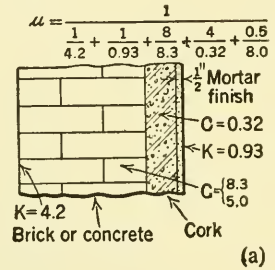
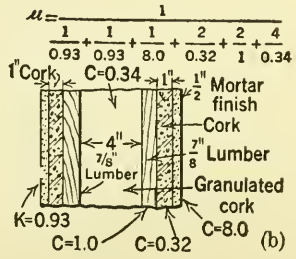


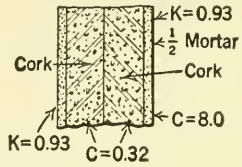

FIG. 134.—Continuous Wall Insulation.

Besides the consideration of waterproofness there are also the factors of fireproofness and protection from vermin. Sawdust has been used in ice-house construction for a long time, but it has nothing to recommend it except its cheapness, and shavings were used in the temporary cold storage buildings erected by the War Department in 1917 and 1918 (Fig. 287). In general, the kind of insulation should conform to the type of construction used in the building, and consideration should be taken of the probable life of the structure as well as the fractional part of the year that the building will probably be used for cold storage or other purposes where insulation is required. While cork is of vegetable origin it can be made waterproof by asphalt coating and to some extent

TABLE 45

| | Thickness of Wall, Inches | Thickness of Cork, Inches | Brick Wall | | | | | | Concrete Wall | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Difference of temperature on the two sides of wall, degrees | | | | | | | | | | | |
| | | | 1 | 10 | 20 | 40 | 60 | 100 | 1 | 10 | 20 | 40 | 60 | 100 |
|   (a) | 8 | 2 | 0.1083 | 1.083 | 2.176 | 4.332 | 6.498 | 10.83 | 0.1164 | 1.164 | 2.328 | 4.656 | 6.984 | 11.64 |
| | | 4 | .0646 | 0.646 | 1.292 | 2.584 | 3.876 | 6.46 | 0.0673 | 0.673 | 1.346 | 2.692 | 4.038 | 6.73 |
| | | 6 | .0460 | 0.460 | 0.92 | 1.840 | 2.760 | 4.60 | 0.0474 | 0.474 | 0.948 | 1.896 | 2.844 | 4.74 |
| | | 8 | .0357 | 0.357 | 0.714 | 1.428 | 2.142 | 3.57 | 0.0366 | 0.366 | 0.732 | 1.464 | 2.196 | 3.66 |
| | 12 | 2 | .0995 | 0.995 | 1.990 | 3.980 | 5.970 | 9.95 | 0.1101 | 1.101 | 2.202 | 4.404 | 6.606 | 11.01 |
| | | 4 | .0614 | 0.614 | 1.228 | 2.456 | 3.684 | 6.14 | 0.0653 | 0.653 | 1.306 | 2.612 | 3.918 | 6.53 |
| | | 6 | .0444 | 0.444 | 0.888 | 1.776 | 2.664 | 4.44 | 0.0464 | 0.464 | 0.928 | 1.856 | 2.784 | 4.64 |
| | | 8 | .0347 | 0.347 | 0.694 | 1.388 | 2.082 | 3.47 | 0.0359 | 0.359 | 0.718 | 1.436 | 2.154 | 3.59 |
| | 16 | 2 | .0923 | 0.923 | 1.846 | 3.692 | 5.538 | 9.23 | 0.1047 | 1.047 | 2.094 | 4.188 | 6.282 | 10.47 |
| | | 4 | .0586 | 0.586 | 1.172 | 2.344 | 3.516 | 5.86 | 0.0633 | 0.633 | 1.266 | 2.532 | 3.798 | 6.33 |
| | | 6 | .0428 | 0.428 | 0.856 | 1.712 | 2.568 | 4.28 | 0.0453 | 0.453 | 0.906 | 1.812 | 2.718 | 4.53 |
| | | 8 | .0338 | 0.338 | 0.676 | 1.352 | 2.028 | 3.38 | 0.0353 | 0.353 | 0.706 | 1.412 | 2.118 | 3.53 |
|  $U = \frac{1}{\frac{1}{0.93} + \frac{1}{0.93} + \frac{1}{8.0} + \frac{2}{0.32} + \frac{2}{1} + \frac{4}{0.34}}$ 1"Cork C=0.34 1" 1/2" Mortar finish Cork 4" 7/8" Lumber 7/8" 7" Lumber Granulated cork K=0.93 C=1.0 C=0.32 C=8.0 (b) | | 2 | .0449 | 0.449 | 0.898 | 1.796 | 2.694 | 4.49 | | | | | | |
| | | 3 | .0394 | 0.394 | 0.788 | 1.576 | 2.364 | 3.94 | | | | | | |
| | | 4 | .0351 | 0.351 | 0.702 | 1.404 | 2.106 | 3.51 | | | | | | |
| | | 6 | .0287 | 0.287 | 0.574 | 1.148 | 1.722 | 2.87 | | | | | | |

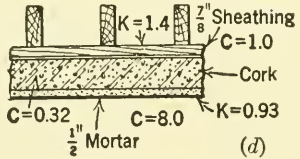
$$u = \frac{1}{\frac{2}{0.93} + \frac{1}{8.0} + \frac{4}{0.32}}$$




(c)

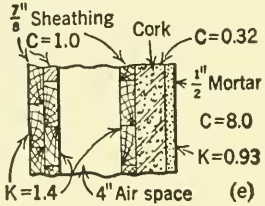
| | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 4 | .0677 | 0.677 | 1.354 | 2.708 | 4.062 | 6.77 | | | | | | |
| 5 | .0559 | 0.559 | 1.118 | 2.236 | 3.354 | 5.59 | | | | | | |
| 6 | .0475 | 0.475 | 0.950 | 1.900 | 2.850 | 4.75 | | | | | | |
| 8 | .0367 | 0.367 | 0.734 | 1.468 | 2.202 | 3.67 | | | | | | |

$$u = \frac{1}{\frac{1}{1.4} + \frac{1}{0.93} + \frac{1}{1} + \frac{0.5}{8.0} + \frac{2}{0.32}}$$




(*d*)

| | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 2 | .110 | 1.10 | 2.20 | 4.40 | 6.60 | 11.0 | | | | | | |
| 4 | .0652 | 0.652 | 1.304 | 2.608 | 3.912 | 6.52 | | | | | | |
| 6 | .0463 | 0.463 | 0.926 | 1.852 | 2.778 | 4.63 | | | | | | |




(e)

$$u = \frac{1}{\frac{1}{0.93} + \frac{3}{1.4} + \frac{3}{1} + \frac{0.5}{8.0} + \frac{2}{0.32}}$$

| | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 2 | .0798 | 0.798 | 1.596 | 3.192 | 4.788 | 7.98 | | | | | | |
| 4 | .0533 | 0.533 | 1.066 | 2.132 | 3.198 | 5.33 | | | | | | |
| 6 | .0400 | 0.400 | 0.800 | 1.600 | 2.400 | 4.00 | | | | | | |

TABLE 46

SPECIFIC CONDUCTIVITY OF VARIOUS MATERIALS

| Materials. | $\alpha_{32}$ | $\beta$ | |
|---|---|---|---|
| Cast iron | 330.00 | −0.0004 | Average values for gray iron. Variations with composition very great. |
| Wrought iron (unworked) | 450.00 | −0.0009 | See next below as indication of variation. |
| Wrought iron (worked) | 240.00 | −0.0006 | |
| Steel (soft) | 300.00 | −0.0003 | See below as indication of variations. |
| Steel (mod. hard) | 240.00 | −0.0003 | |
| Steel (very hard) | 180 | ..... | |
| Copper (pure) | 2400 | −0.0002 | Values given by different experimenters vary considerably. Probably due to variations in purity and condition. |
| Copper (commercial) | 2100. | +0.0001 | |
| Brass (yellow) | 420 | +0.0014 | Varies greatly with composition. |
| Brass (red) | 540 | +0.0008 | |
| Aluminum (pure) | 750 | +0.0003 | |
| Aluminum with { 0.5% Fe, 0.4% Cu } | 1041 | +0.0002 | |
| Cylinder oil | 0.784 | −0.0015 | Naturally varies with kind of oil, cylinder oil not being a definite compound. |
| Water | 2.6158 | +0.0053 | These values seem best authenticated. Authorities differ greatly. |
| Air | 0.11989 | +0.0017 | Varies with humidity, etc. |
| Hydrogen | 0.71933 | ..... | |
| Fire brick | ..... | ..... | 6.948 at 1300° F. Varies considerably with composition of brick. |
| "Insulating" materials | 0.4 to 1.2 | ..... | Such materials as cork, cellular paper, asbestos mixtures, etc. |

$$\alpha_t = \alpha_{32}\{1 + \beta\,(t-32)\}.$$

## TABLE 47

VALUES OF $r_2 \log_e (r_2/r_1)$ FOR COVERINGS OF VARIOUS THICKNESSES ON DIFFERENT-SIZED PIPES

| Pipe Size, Inches | Thickness of Covering | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1 in. | | | 1½ in. | | 2 in. | | 2½ in. | | 3 in. | | 4 in. | |
| | $r_1$ | $r_2$ | $r_2 \log_e \frac{r_2}{r_1}$ | $r_2$ | $r_2 \log_e \frac{r_2}{r_1}$ | $r_2$ | $r_2 \log_e \frac{r_2}{r_1}$ | $r_2$ | $r_2 \log_e \frac{r_2}{r_1}$ | $r_2$ | $r_2 \log_e \frac{r_2}{r_1}$ | $r_2$ | $r_2 \log_e \frac{r_2}{r_1}$ |
| ⅛ | 0.420 | 1.420 | 1.730 | 1.920 | 2.920 | 2.420 | 4.235 | 2.920 | 5.660 | 3.420 | 7.165 | 4.420 | 10.400 |
| ¼ | 0.525 | 1.525 | 1.625 | 2.025 | 2.733 | 2.525 | 3.965 | 3.025 | 5.294 | 3.525 | 6.710 | 4.525 | 9.760 |
| 1 | 0.657 | 1.657 | 1.532 | 2.157 | 2.563 | 2.657 | 3.710 | 3.157 | 4.950 | 3.657 | 6.278 | 4.657 | 9.110 |
| 1¼ | 0.830 | 1.830 | 1.448 | 2.330 | 2.402 | 2.830 | 3.470 | 3.330 | 4.622 | 3.830 | 5.855 | 4.830 | 8.500 |
| 1½ | 0.950 | 1.950 | 1.400 | 2.450 | 2.322 | 2.950 | 3.340 | 3.450 | 4.445 | 3.950 | 5.620 | 4.950 | 8.170 |
| 2 | 1.187 | 2.187 | 1.335 | 2.687 | 2.194 | 3.187 | 3.145 | 3.687 | 4.177 | 4.187 | 5.262 | 5.187 | 7.658 |
| 2½ | 1.437 | 2.437 | 1.288 | 2.937 | 2.062 | 3.437 | 2.996 | 3.937 | 3.965 | 4.437 | 5.000 | 5.437 | 7.230 |
| 3 | 1.750 | 2.750 | 1.240 | 3.250 | 2.010 | 3.750 | 2.856 | 4.250 | 3.770 | 4.750 | 4.740 | 5.750 | 6.840 |
| 3½ | 2.000 | 3.000 | 1.216 | 3.500 | 1.959 | 4.000 | 2.770 | 4.500 | 3.647 | 5.000 | 4.580 | 6.000 | 6.590 |
| 4 | 2.250 | 3.250 | 1.190 | 3.750 | 1.913 | 4.250 | 2.700 | 4.750 | 3.598 | 5.250 | 4.450 | 6.250 | 6.385 |
| 4½ | 2.500 | 3.500 | 1.177 | 4.000 | 1.880 | 4.500 | 2.640 | 5.000 | 3.465 | 5.500 | 4.340 | 6.500 | 6.210 |
| 5 | 2.781 | 3.781 | 1.161 | 4.281 | 1.845 | 4.781 | 2.588 | 5.281 | 3.383 | 5.781 | 4.226 | 6.781 | 6.043 |
| 6 | 3.312 | 4.312 | 1.138 | 4.812 | 1.795 | 5.312 | 2.507 | 5.812 | 3.266 | 6.312 | 4.071 | 7.312 | 5.790 |
| 7 | 3.812 | 4.812 | 1.117 | 5.312 | 1.761 | 5.812 | 2.448 | 6.312 | 3.190 | 6.812 | 3.955 | 7.812 | 5.626 |
| 8 | 4.312 | 5.312 | 1.101 | 5.812 | 1.732 | 6.312 | 2.405 | 6.812 | 3.114 | 7.312 | 3.860 | 8.312 | 5.450 |
| 9 | 4.812 | 5.812 | 1.109 | 6.312 | 1.710 | 6.812 | 2.370 | 7.312 | 3.060 | 7.812 | 3.790 | 8.812 | 5.335 |
| 10 | 5.375 | 6.375 | 1.085 | 6.875 | 1.694 | 7.375 | 2.334 | 7.875 | 3.007 | 8.375 | 3.707 | 9.375 | 5.220 |
| 12 | 6.370 | 7.375 | 1.073 | 7.875 | 1.660 | 8.375 | 2.285 | 8.875 | 2.932 | 9.375 | 3.614 | 10.375 | 5.060 |
| 14 | 7.000 | 8.000 | 1.067 | 8.500 | 1.654 | 9.000 | 2.262 | 9.500 | 2.900 | 10.000 | 3.566 | 11.000 | 4.986 |
| 16 | 8.000 | 9.000 | 1.059 | 9.500 | 1.630 | 10.000 | 2.236 | 10.500 | 2.852 | 11.000 | 3.500 | 12.000 | 4.860 |
| 18 | 9.000 | 10.000 | 1.048 | 10.500 | 1.620 | 11.000 | 2.203 | 11.500 | 2.817 | 12.000 | 3.455 | 13.000 | 4.795 |

$$Q = \frac{t_1 - t_2}{\frac{r_s \log_e \frac{r_2}{r_1}}{C_1} + \frac{r_s \log_e \frac{r_3}{r_2}}{C_2} + \frac{r_s \log_e \frac{r_4}{r_3}}{C_3} +}$$

B.t.u. per square foot of outside area.

by the mortar finish. The mortar finish assists in making cork fireproof, but under any conditions it is a very slow burning material. At the present time corkboard is used in the majority of installations where flat surfaces are to be insulated. In the case of tanks, shell and tube brine coolers, etc., molded insulation can be used to advantage, but in many cases granulated cork can be packed around the body to be insulated and held in place by means of a container made of tongue and grooved lumber with waterproof paper between the layers. A good example of this construction is shown in Fig. 270 for ice tank construction.

TABLE 48

FOR DRINKING WATER CONDITIONS

Heat leakage in B.t.u. per linear foot of pipe per deg. F. difference of temperature per hour

| Size of Pipe | Union Lith | Hair Felt (Two Layers) | Cork-board | Size of Pipe | Union Lith | Hair Felt (Two Layers) | Cork-board |
|---|---|---|---|---|---|---|---|
| $\frac{1}{2}$ | 0.157 | 0.086 | 0.160 | 3 | 0.294 | 0.192 | 0.304 |
| $\frac{3}{4}$ | 0.170 | 0.095 | 0.167 | $3\frac{1}{2}$ | 0.355 | 0.211 | 0.330 |
| 1 | 0.180 | 0.107 | 0.178 | 4 | 0.367 | 0.228 | 0.345 |
| $1\frac{1}{4}$ | 0.213 | 0.124 | 0.199 | $4\frac{1}{2}$ | 0.376 | 0.247 | 0.385 |
| $1\frac{1}{2}$ | 0.217 | 0.132 | 0.220 | 5 | 0.446 | 0.268 | 0.410 |
| 2 | 0.269 | 0.149 | 0.245 | 6 | 0.558 | 0.307 | 0.437 |
| $2\frac{1}{2}$ | 0.272 | 0.167 | 0.291 | | | | |

The use of corkboard for pipe line construction is not always the best practice. Where the pipe has a tendency to shake, the material should have the ability to give a little. Under these conditions hair felt gives satisfaction if properly applied, that is, if the material is kept from absorbing moisture as it will unless the outside waterproofing is erected in place correctly.

**The Effect of Moisture.**—The effect of moisture on the value of the coefficient of heat transfer is very well explained in the paper by L. F. Miller presented at the 14th Western meeting of the American Society of Refrigerating Engineers, May, 1927. In this paper it is shown by experiments that the increase in the value of $k$ may be as much as 50 per cent. It is also mentioned in the paper, a fact that others have noticed, that the value of $k$ varies with the actual temperature of the material subject to heat transmission. As a rule the refrigerating

engineer is not particularly interested in either of these factors. The insulation is waterproofed as far as possible and the engineer realizes that there is a tendency for moisture to work into the material where freezing will ultimately disintegrate it. The usual practice is to make the insulation waterproof, or as nearly so as is practical, and to use a factor in the calculations for heat leakage that will allow for irregularities in the erection and for the presence of moisture.

**Erection.**—Dead air spaces in the floor or the wall are of no value in cold temperature insulation, but are a source of considerable trouble due to condensate collecting in these spaces, as is found in hollow tile or the space between joists. Roof and floor slabs are preferred to tile, and where air spaces are found in the construction it is best to fill them with granulated cork or other insulating material. Care must be taken in the insulation of columns or tanks on the ground floor or freezing of the earth under the footings will result. Where cork is applied to a concrete ceiling it may be laid in the concrete forms, and a finish of portland cement mortar can be applied after the forms have been removed. The larger cold storage warehouses are now constructed with continuous outside wall insulation without break due to tying the floors to the walls (Fig. 144). In rooms where the humidity is high, and where there is a tendency for moisture to condense on the surface, the cork should be covered with some special preparation. The Armstrong Cork Co. uses an *asphalt mastic* finish $\frac{1}{8}$ in. thick, which is ironed on at the factory and is made tight at the joints on erection by means of a hot trowel. If nails are used in corkboard they should be galvanized, but some engineers prefer the wooden skewer as wood is a poorer conductor of heat. Pitch or tar should not be used in the erection of corkboard, but instead some odorless material which will not tend to taint the foodstuffs in storage. Other things being equal, the asphalt is better than the cement plaster because of the waterproofness of the asphalt. Moisture will penetrate any form of insulation if the conditions are favorable.

In erecting corkboard against brick or concrete walls the first process should be to prime the masonry thoroughly with at least one coat of some sort of emulsified asphalt primer. If such a priming coat is not applied to the walls, and the corkboard is erected against the walls in portland cement, the air which seeps through will carry moisture with it which may condense back of the corkboard and even enter the interstices between the granules of cork, thereby causing a disintegration of the cork.

The use of asphalt or cement plaster in cold storage construction is still not definitely settled. Cement gives greater strength than asphalt

does, but the latter is waterproof and costs considerably less than cement mortar. Cement plaster will crack, therefore the plaster finish should be applied in two coats and score marks should be made every four feet.

**Protecting and Finishing Cork Insulation.**—A finish for cork insulation, in order to have a universal application to all conditions under which such insulation is used, should have the following properties:

1. It should bond securely to the face of the cork under all conditions of service and use.

2. It should remain intact—that is, should not develop cracks regardless of the temperature changes to which it is subjected. In other words, the continuity of the finish should not be broken.

3. It should be both air-proof and moisture-proof in order to insure the maximum insulation value of the cork sheet.

4. It should be sanitary—not absorbent of odors—and it should have a finish that may be cleaned readily by steam or hot water without damage.

5. It should be odorless itself, and in setting it should give off no fumes or noxious odors, as it must be used frequently in confined spaces.

6. It should be of such a character as to take decoration, as a paint or an enamel, should a paint finish be desired.

7. Its application should be within the range of the abilities of one of the usual building trades without special equipment or special knowledge.

8. It must be economical.

For the great majority of work there has been no alternative in the selection of the finish. The choice has been limited to the ordinary plaster, not because it was satisfactory and fully met service conditions, but because it was the only material available.

Plaster, however, is objectionable, and has proved unsatisfactory in many particulars. In the first place it does not bond well with cork. Cork is without suction or capillarity, and a surface that lacks this property is difficult to plaster on with any assurance of good adhesion. There are many instances where plaster has fallen off the cork sheets, and has had to be replaced with the addition of wire mesh or expanded metal in order to hold it in place.

Plaster not only develops cracks, but the very conditions of its use make the development of cracks inevitable. Cork sheets are almost always set up in hot asphalt, which is used as a cementing medium, although asphalt is not a a solid but is in reality a fluid. When the wall against which the cork is laid expands or contracts, some slight movement of the cork takes place. When this occurs the rigid plaster on the

face of the cork generally yields by cracking. It is claimed also that plaster is neither air-proof, moisture-proof nor sanitary since it will absorb both moisture and odors.

**Standard Construction.**—The following specifications and construction[4] represent good practice for pipes, fittings, tanks and building construction.

**Insulation of Refrigerated Piping.**—After all the pipe lines have been tested and have been made tight, they should be covered with cork covering and fitting covers of the proper thickness, as outlined in the following schedules:

| | |
|---|---|
| For temperatures from 0 deg. F. to 25 deg. F. | Brine thickness. |
| For temperatures from 0 deg. F. to − 35 deg. F. | Special thick brine. |
| For temperatures from 25 deg. F. to 45 deg. F. | Ice water thickness. |

All foreign matter, such as plaster, should be removed from all pipe surfaces before covering is applied. Under no circumstance shall pipes be covered while wet or in a frosted condition.

Sectional covering shall be used on all pipes 8 in. and smaller for brine thicknesses, 6 in. and smaller for special thick brine, and 10 in. and smaller for ice water thickness.

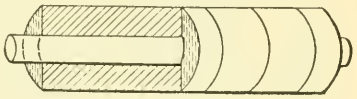


Fig. 135.—Pipe Insulation.

Sectional covering shall be applied with all end joints broken by making one-half of the first section 18 in. long and leaving the other half 36 in. in length (Fig 135). All longitudinal joints shall be on the top and the bottom of the pipe and not along the sides. All covering shall be applied with waterproof cement on all joints and wired in place with copper-clad steel wire, using not less than six wires to a section.

For brine thickness molded fitting covers shall be used on all screwed and flanged fittings 6 in. and smaller. In the case of special thick brine, molded fitting covers shall be used on standard screwed fittings 5 in. and smaller, ammonia and extra heavy screwed fittings $4\frac{1}{2}$ in. and smaller, and all flanged fittings 4 in. and smaller. For ice water thickness molded fitting covers shall be used on standard and extra heavy screwed fittings 6 in. and smaller, standard flanged fittings 4 in. and smaller.

Molded fitting covers shall be applied with waterproof cement on all joints and wired in place with copper-clad steel wire, using not less than four wires on screwed fittings and six wires in flanged fittings

[4] The following covers the Armstrong Cork and Insulating Co. practice.

(Figs. 136 and 137). All spaces between covers and fittings shall be filled carefully with brine putty.

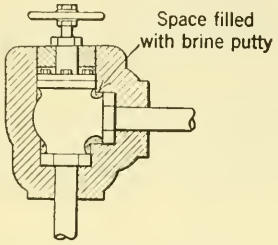



Fig. 136.—Insulation for an Angle Valve.

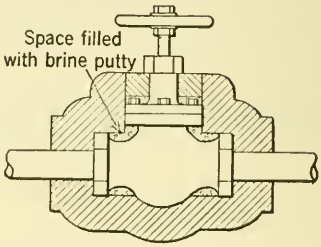



Fig. 137.—Insulation for a Globe Valve.

After the covering has been applied, all seams and chipped edges shall be filled with a suitable seam filler so as to leave a smooth and even surface. The whole covering shall then be given one good coat of cork covering paint.

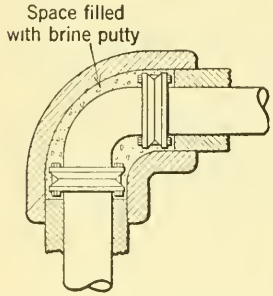



Fig. 138.—Insulation—Flanged Ell.

All pipes larger than the sizes listed for sectional covering shall be covered with cork lagging weighing not less than 1.2 lb. per foot, board measure, and beveled to the proper diameter. All joints shall be broken by starting with alternate 18 ins. and 36 in. lags. Waterproof cement shall be used on all joints, and the lags shall be wired in place with copper-clad steel wire using not less than six wires to every three feet of lagging.

Screwed and flanged fittings larger than the sizes just mentioned for molded covers shall be covered with fitting covers made from cork

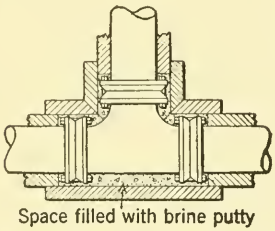



Fig. 139.—Insulation—Flanged Tee.

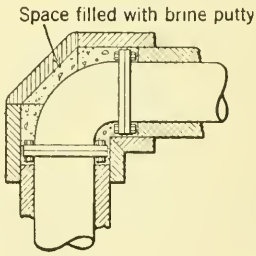



Fig. 140.—Insulation Flanged Ell of Large Size, with Cover.

lagging weighing not less than 1.2 lb. per foot, board measure, and beveled to the proper diameter. All lagging shall be assembled with waterproof cement on all joints and wired in place with copper-clad steel wire, using not less than six wires per fitting (Figs. 138 and 140).

**Cylindrical Brine Coolers and Tanks.**—The following table gives the thickness of cork lagging that has been found economical for the temperatures noted:

| | Inches |
|---|---|
| Below 5 deg. F. | 6 |
| 5 deg. to 20 deg. F. | 5 |
| 20 deg. to 32 deg. F. | 4 |
| 32 deg. to 55 deg. F. | 3 |
| 55 deg. to 65 deg. F. | 2 |

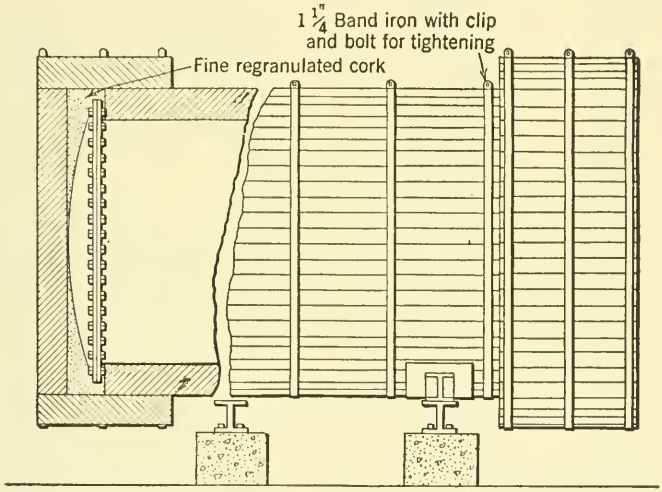



Fig. 141.—Insulation—Horizontal Brine Cooler.

The cylindrical surface of the tank will be insulated with one layer of cork lagging of the proper thickness. The lagging shall be beveled to the proper diameter and shall have a mineral rubber finish ironed on at the factory to both the inner and the outer surfaces. The lags shall be applied with waterproof cement on all joints and secured with $1\frac{1}{4}$-in. bands of 16-gage galvanized iron, drawn up tight by means of bolts through clips riveted to the ends of the bands. The bands shall be spaced not more than 12 in. apart. The lags shall be applied with broken transverse joints. All spacing between the tank and the lagging shall be filled with brine putty.

The flanged ends of the cooler or the tank shall be insulated with a suitable disk of cork lagging supported by a layer of cork lagging over

the flanges. This flange lagging shall project beyond the head of the tank a distance equal to the thickness of the disk and shall have a bearing on the body lagging of not less than the overhand required to enclose the disk, tank head and flange bolts. The flange lagging shall be applied with waterproof cement on all joints and secured by not less than three galvanized iron bands as described.

**Walls.**—It is recommended that the walls of ice storage rooms be built of hard pressed brick. The roof must be left around all walls for a

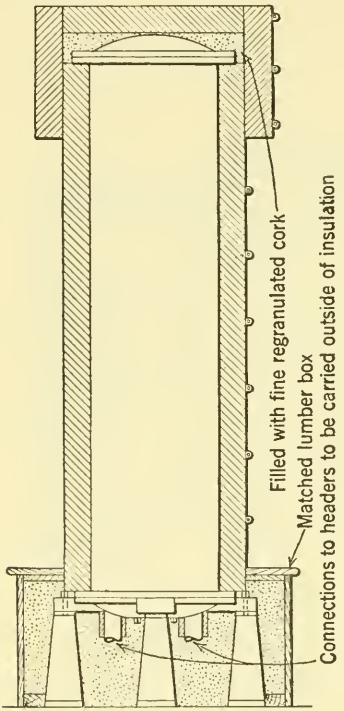



Fig. 142.—Insulation—Vertical Brine Cooler.

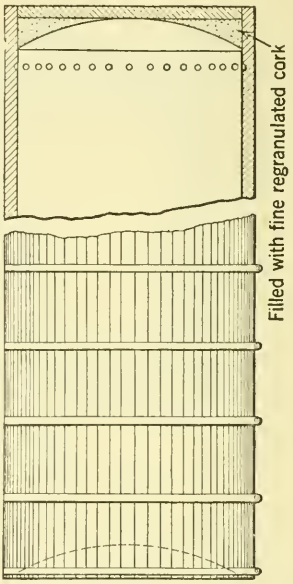



Fig. 143.—Insulation—Riveted Brine Cooler.

space equal to the thickness of the wall insulation plus 1 in., so that the wall insulation may be continuous to its connection with the roof insulation, as shown in Fig. 145. Any spaces between the insulation and the edge of the roof slab shall be filled with cement grout.

The wall surfaces must be smooth and true, and thoroughly cleaned of all dirt, dust, plaster, loose mortar, etc. If rough and uneven, the wall surfaces must be plastered with portland cement plaster, mixed one part of cement to three parts of clean sand, applied thick enough to

true up the wall surface and not less than $\frac{1}{4}$ in. thick at the high points. The surfaces of the walls (if plastered, after the plaster has dried thoroughly) must then be painted with a good coat of asphalt paint.

Prepare a suitable amount of asphalt cement. Use plain unscored corkboard. Dip each piece—one side, the end that is to butt the board that is already in place, and the lower edge—in asphalt, then put it in place quickly. The first row of corkboard must be erected on the wall at the floor on a level line, and should be carried clear around the room if all four walls are to be insulated. The boards must be kept in perfect alignment, in order that the joints in the upper rows may fit close and tight. As each board is put in place care must be taken to butt it tightly against the adjoining boards. All vertical joints must be broken, or must be staggered, as shown in Fig. 145.

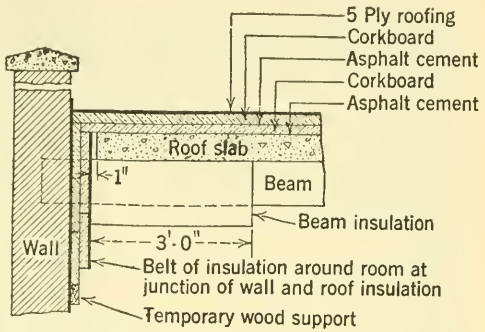



Fig. 144.—Insulation—Wall and Floor.

A second course shall be laid as follows: Saw some of the boards lengthwise down the center, so as to have enough pieces of half the regular width to make one row around the room. Start at the floor line with these narrow pieces, staggering joints with the floor insulation as shown in Fig. 144. Dip each piece—the plain side, the end that is to abut the board that is already laid in place, and the lower edge—in asphalt, and then put it in place quickly, drawing it up tightly against the first course of corkboard with either galvanized iron nails or wooden skewers. Care must be taken to prevent the vertical joints of the second course from coming directly over the vertical joints in the first course. Then continue the erection of the second

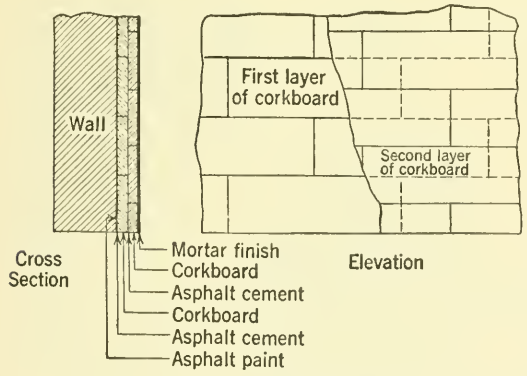



Fig. 145.—Insulation—Wall.

layer of corkboard, using sheets of full length and width. Care must be taken to keep the first row in perfect alignment so that the upper rows may be close and tight, as the efficiency of the insulation depends upon getting tight joints. At the floor and roof and in the corners of the room the two layers of corkboard should be joined in the manner shown in Fig. 146.

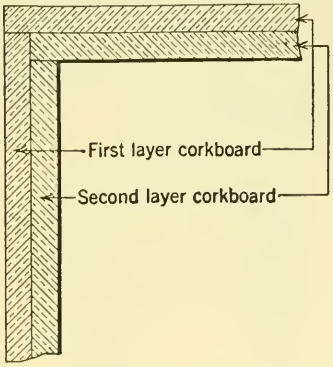



Fig. 146.—Insulation—Joint at Wall and Ceiling.

The nails or skewers should be driven in obliquely, two to the square foot. They must be long enough to reach well into the first course, but not through the wall. Two nails or skewers, one near each end, should be driven part way into each board before it is put in place and used to hold the board when dipping it in the asphalt cement.

All joints and nail heads shall be filled with a plastic seam filler. Care must be taken to see that the joints between the wall and the roof insulation are staggered and made perfectly tight by sealing with hot asphalt in the manner shown in Fig. 146. In case the roof insulation is to be laid before the walls are insulated, a belt of insulation must first be erected around the top of the walls in order to insure the proper connection of wall and roof insulation. This belt should be wide enough to extend slightly below the roof beams and should rest on a wood support temporarily secured to the wall, as shown in Fig. 146, until the wall insulation is put up.

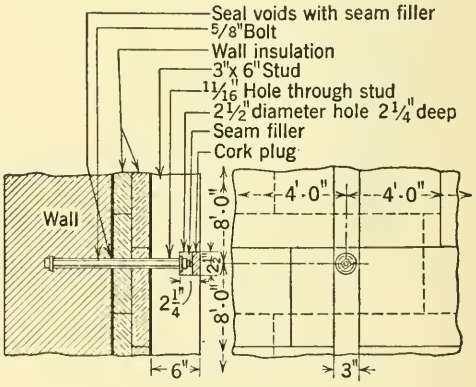



Fig. 147.—Insulation—Buckstay.

Where the height of the room is over 14 ft., *buckstays*, or studding, 3 × 6 in., must be erected on 4-ft. centers horizontally and 8-ft centers vertically to a depth of 8 in. in the wall and extending through the insulation and approximately 4 in. beyond. Any voids around the bolts at the face of the brick wall shall be

packed with seam filler before the corkboard is erected. The buck-stays will fasten securely 3 × 6 studs erected on edge and drawn up tight against the insulation, as shown in Fig. 147.

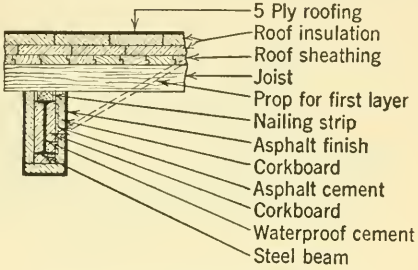



Fig. 148.—Insulation—Steel Beams.

**Beams.**—The *steel beams* supporting the roof must be insulated for a distance of 3 ft. out from the wall line with the same thickness of insulation used on the walls. The first layer is to be erected against the beam with waterproof cement and propped in place until the cement sets, as shown in Fig. 148. *Concrete beams* are to be insulated in the same manner as the wall and for the same distance as specified for steel beams.

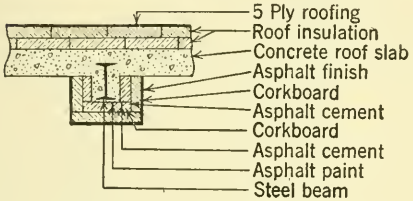



Fig. 149.—Insulation—Concrete Beams.

**Roof.**—The concrete or wood roof, upon which the corkboard is to be applied, must first be cleaned. All dirt, plaster, dust, loose mortar, whitewash, etc., must be removed carefully. In the case of a concrete slab, if the surface is not reasonably smooth, it should be covered with a light coating of portland cement in order to make an even surface against which to apply the corkboard. Prepare a suitable amount of asphalt on the basis of approximately two and one-quarter pounds for each square foot of roof surface.

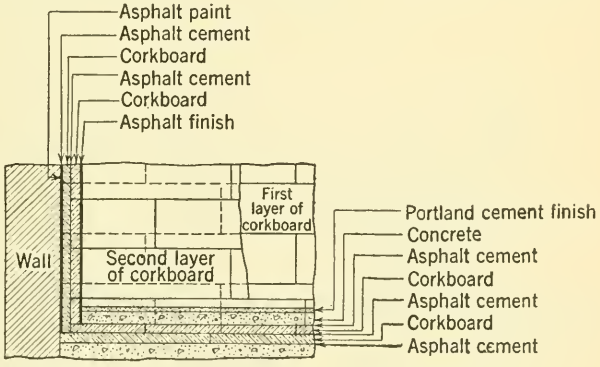



Fig. 150.—Insulation—Wall and Floor.

Use plain corkboard; that is, corkboard that has not been scored. Pour out on the roof only sufficient asphalt to lay one board at a time. As each sheet is put down push it along a few inches in the melted asphalt so that sufficient asphalt

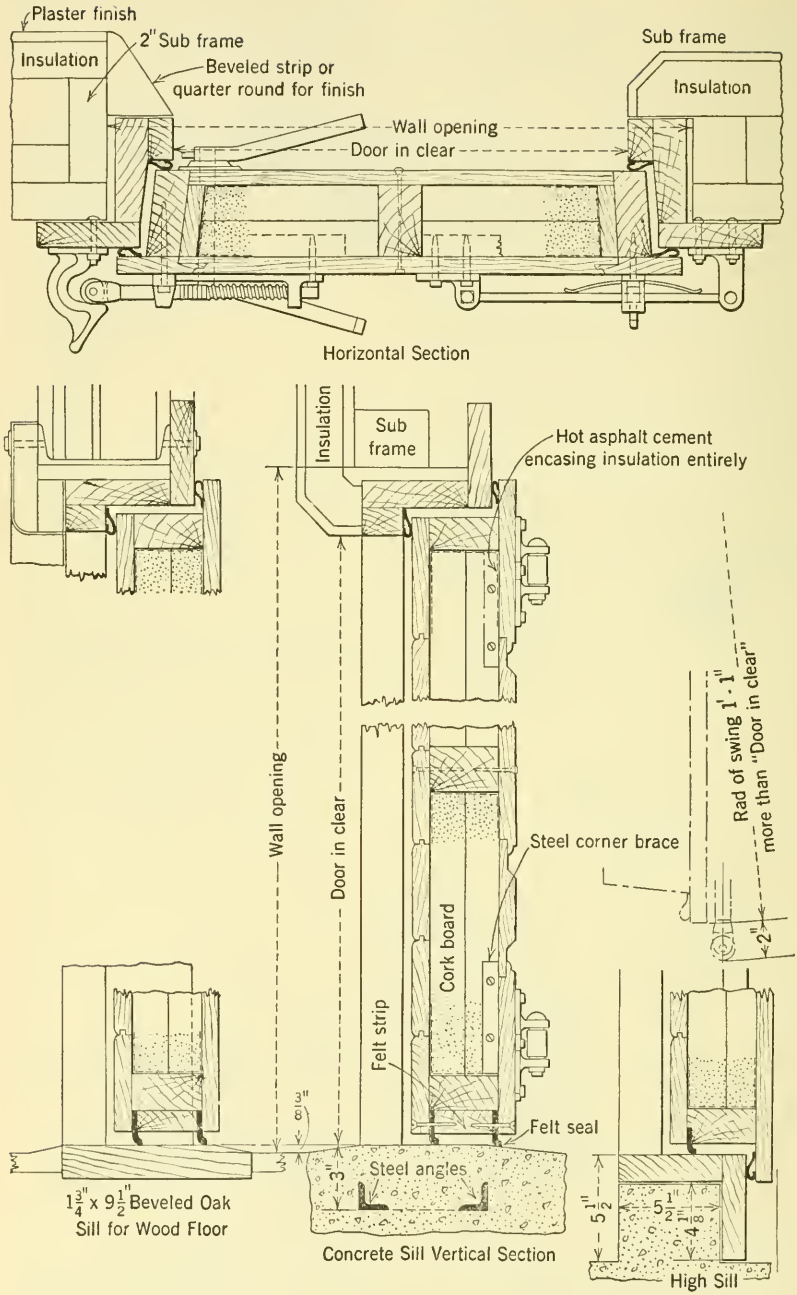



FIG. 151.—Insulation—Cold Storage Door.

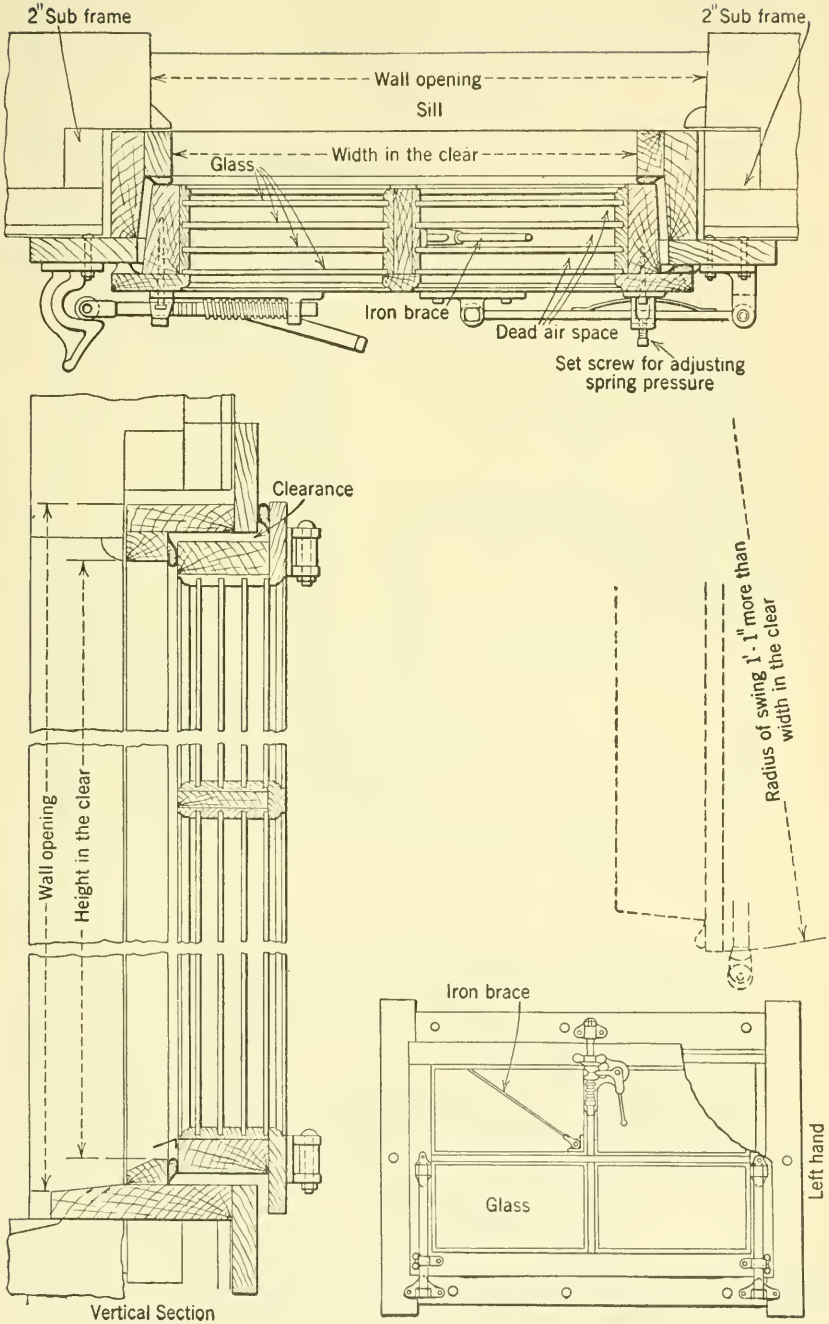



Fig. 152.—Insulation—Cold Storage Window.

will accumulate on the edge of the board to seal the joints effectively. Put down the first row along one side of the roof. Be careful to stagger

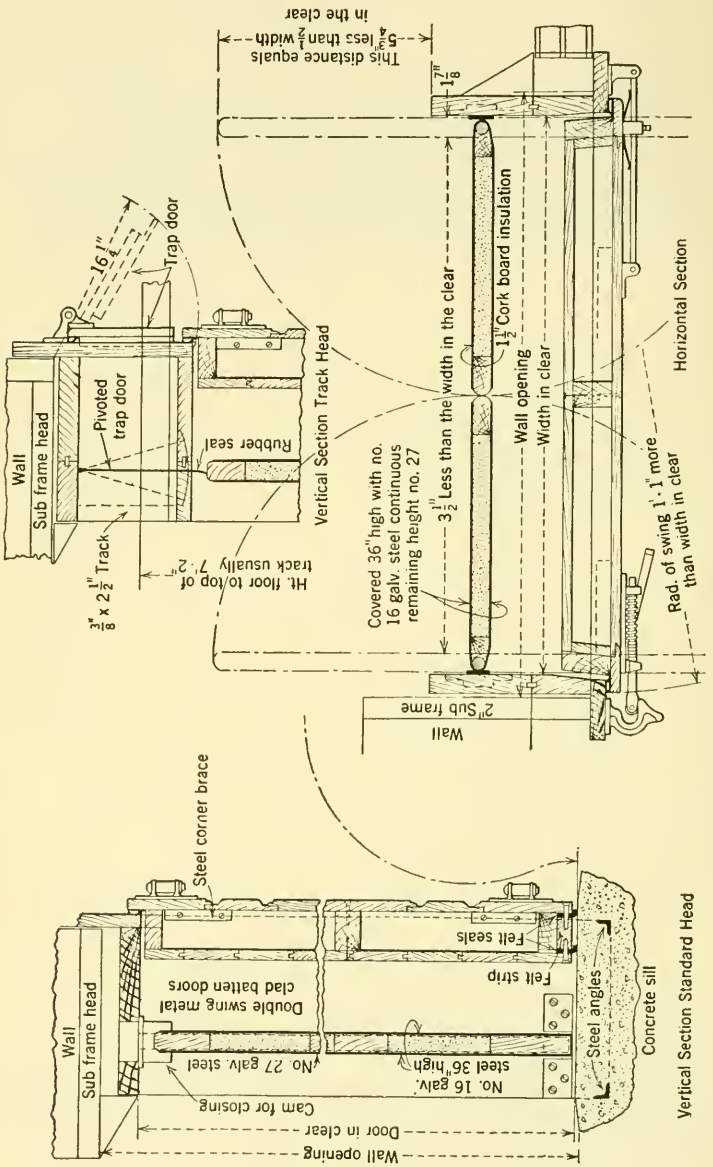



FIG. 153.—Insulation—Cold Storage Door.

joints with the wall insulation and to seal these contacts with hot asphalt. Take care to keep the boards in perfect alignment, so as to

secure tight fitting joints in the rows to follow. Break the joints in the second row and the second course in a manner similar to that used in wall construction, but be sure that the short joints of the second course do not come over the short joints of the first course. Especial care must be taken to insure that the joint between the roof and the wall insulation is perfectly tight and well sealed with asphalt.

**Floor.**—The corkboard shall be applied in a manner similar to that used in roof and wall insulation, after which the surface shall be flooded with hot asphalt to a thickness of at least $\frac{1}{8}$ in. by simply pouring the

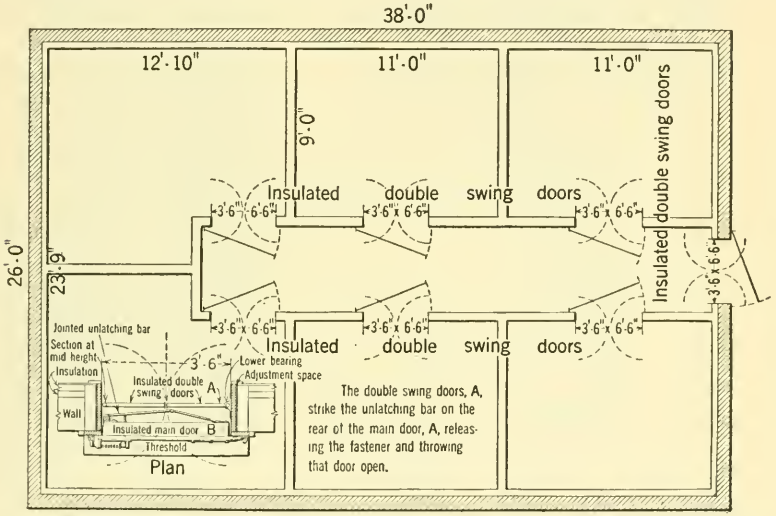



FIG. 154.—Cold Storage Doors—The Double Swing Door.

molten asphalt uniformly over the surface. On this surface a 4-in. concrete wearing floor shall be laid down.

Tables 50 and 51 give dimensions and shipping weights for corkboard. The design and construction of cold storage doors are shown in Figs. 151, 153 and 154, and a typical cold storage window by Fig. 152. Bunker room construction, showing the opening for the warm and the cold air, and the insulation required are shown in Figs. 155 and 156.

**The Economic Thickness of Insulation.**—It has been the custom of engineers to use the rough rule of 1 in. of corkboard for every 10 deg. F. difference in temperature between that of the cold storage room or warehouse and the mean temperature of the atmosphere. This usually resulted in thicknesses of insulation of from 3 to 4 in. in the case of cooler walls and 6 to 9 in. for freezer rooms, depending on the extreme limits to be experienced in the work. Like most rules of thumb, the thickness

to be used has no reference to the cost of the power consumed or of the investment either in the building or in the equipment directly con-

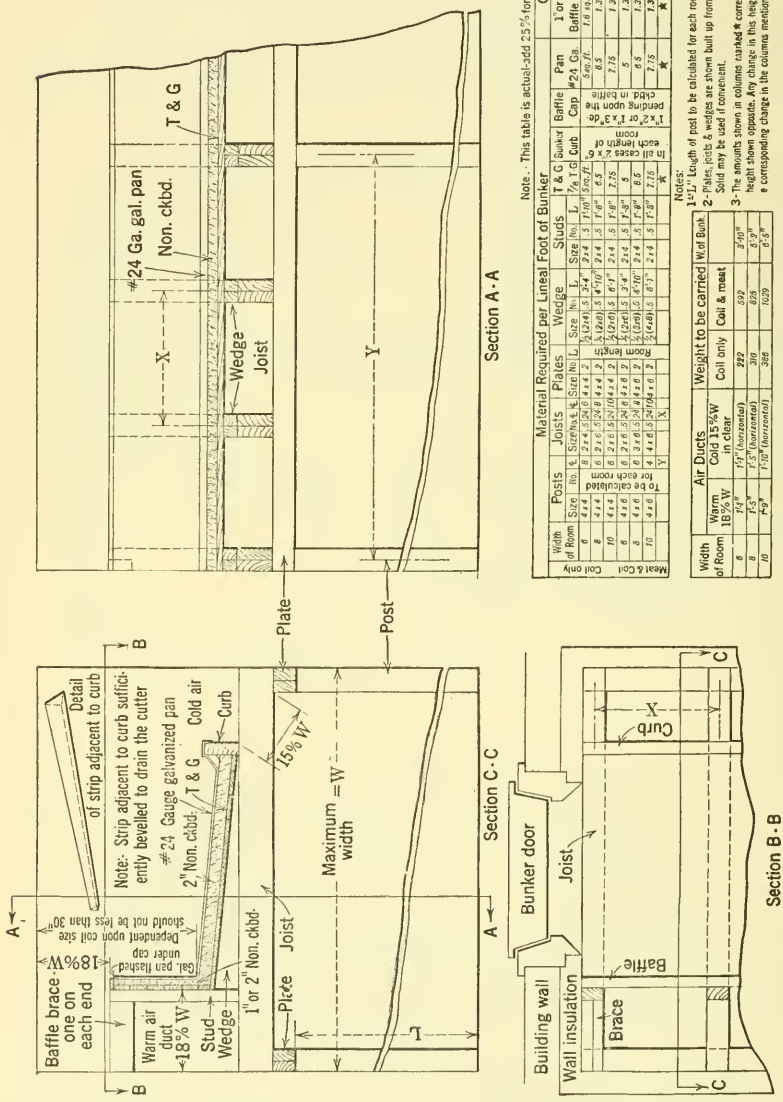
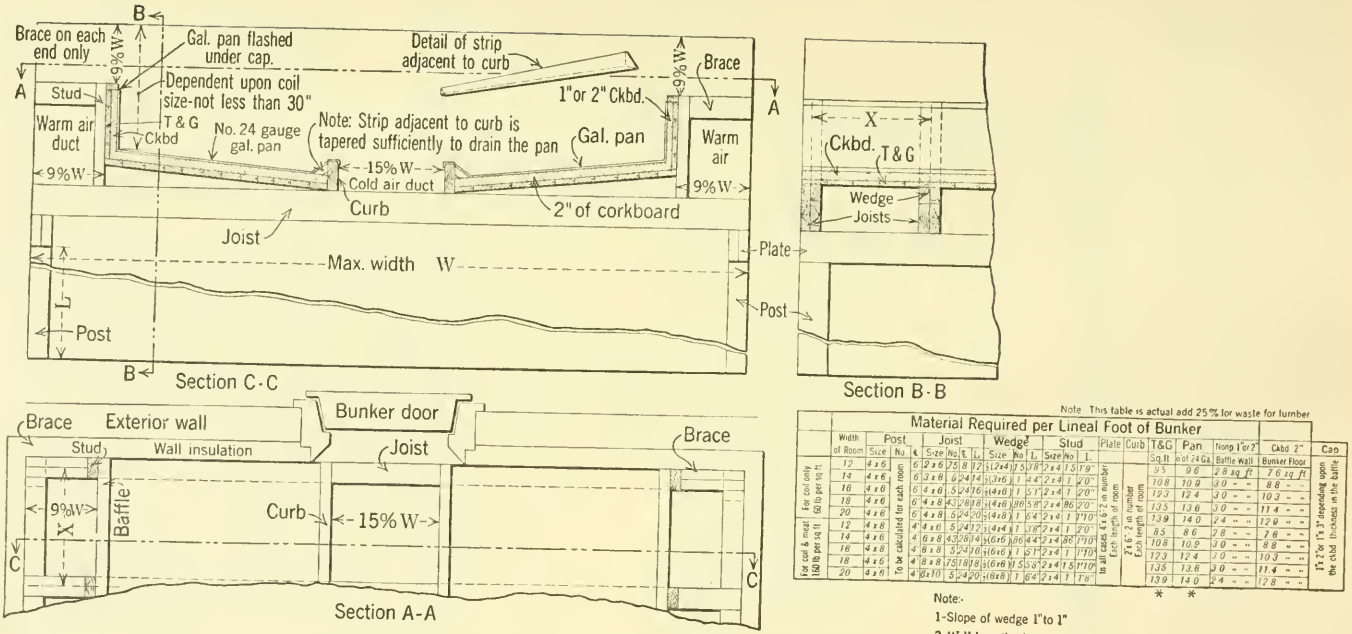



FIG. 155.—Insulation—Single Bunker.

cerned in the production of refrigeration. It is evident, when one considers the subject, that if a ton of refrigeration costs when delivered to




Note This table is actual add 25% for waste for lumber

Material Required per Lineal Foot of Bunker

| | Width of Room | Post Size | Post No | Joist Size | Joist No | Joist t | Joist L | Wedge Size | Wedge No | Wedge L | Stud Size | Stud No | Stud L | Plate | Curb | T&G Sq. ft | Pan lot 24 Ga | Nonp 1"or2" Battle Wall | Cabd 2" Bunker Floor | Cap |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| For coil only (50 lb per sq ft | 12 | 4 x 6 | 6 | 2 x 6 | 7.5 | 8 | 12 | 1 (2x4) | 1 | 5 | 2 x 4 | 1 | 5'10" | | | 9.5 | 9.6 | 2.8 sq ft | 7.6 sq ft | |
| | 14 | 4 x 6 | 6 | 3 x 6 | 6 | 24 | 14 | 1 (3x6) | 1 | 4.4 | 2 x 4 | 1 | 6'0" | | | 10.8 | 10.9 | 3.0 " " | 8.8 " " | |
| | 16 | 4 x 6 | 6 | 4 x 6 | 5.2 | 4 | 16 | 1 (4x6) | 1 | 3.7 | 2 x 4 | 1 | 7'0" | | | 12.3 | 12.4 | 3.0 " " | 10.3 " " | |
| | 18 | 4 x 6 | 6 | 4 x 8 | 4.3 | 6 | 18 | 1 (4x6) | 1 | 3.26 | 2 x 4 | 1 | 8'0" | | | 13.5 | 13.6 | 3.0 " " | 11.4 " " | |
| | 20 | 4 x 6 | 6 | 4 x 8 | 5.2 | 4 | 20 | 1 (4x8) | 1 | 6.4 | 2 x 4 | 1 | 9'10" | | | 13.9 | 14.0 | 2.4 " " | 12.0 " " | |
| For coil & meat (60 lb per sq ft | 12 | 4 x 8 | 4 | 6 x 8 | 4.2 | 6 | 14 | 1 (4x4) | 1 | 3.78 | 2 x 4 | 1 | 7'0" | | | 8.5 | 8.6 | 2.8 " " | 7.6 " " | |
| | 14 | 4 x 6 | 4 | 6 x 8 | 4.2 | 6 | 14 | 1 (6x6) | 1 | 4.4 | 2 x 4 | 1 | 7'0" | | | 10.8 | 10.8 | 3.0 " " | 8.8 " " | |
| | 16 | 4 x 8 | 4 | 6 x 8 | 5.2 | 4 | 16 | 1 (6x6) | 1 | 3.7 | 2 x 4 | 1 | 7'0" | | | 12.3 | 12.4 | 3.0 " " | 10.3 " " | |
| | 18 | 4 x 6 | 4 | 8 x 8 | 7.5 | 1 | 8 | 1 (6x6) | 1 | 5.5 | 2 x 4 | 1.5 | 7'10" | | | 13.5 | 13.6 | 3.0 " " | 11.4 " " | |
| | 20 | 4 x 6 | 4 | 8 x 10 | 5.2 | 4 | 20 | 1 (6x8) | 1 | 6.4 | 2 x 4 | 1 | 9'0" | | | 13.9 | 14.0 | 2.4 " " | 12.8 " " | |

To be calculated for each room

Add 1/4 to the number for each length of room

2'6" in number Each length of room

1'3" or 1'9" depending upon the cbd thickness in the baffle

* *

Note:-

1-Slope of wedge 1" to 1"

2-"L" Length of posts to be calculated for each room

3-Plates, joists, and wedges are shown built up from 2"&3" material. Solid may be used it convenient.

4-The amounts shown in columns marked "*" corresponds to the stud height shown opposite. Any change in this height will necessitate a corresponding change in the amounts mentioned above.

| Width of Room | Size of Air Duct 18%W Warm Air | | 15%W Cold Air | Width of Bunker | | Total Load per Lin. Ft Coils only | Meat & Coil |
|---|---|---|---|---|---|---|---|
| W | Total | Each | | Total | Each | | |
| 12 | 2'-2" | 1'-1" | 1'-10" | 8'-0" | 4'-0" | 480 | 1280 |
| 14 | 2'-6" | 1'-8" | 2'-2" | 9'-4" | 4'-8" | 680 | 1494 |
| 16 | 2'-10" | 1'-5" | 2'-4" | 10'-0" | 5'-5" | 850 | 1732 |
| 18 | 3'-4" | 1'-8" | 2'-8" | 12'-0" | 6'-0" | 770 | 1920 |
| 20 | 3'-8" | 1'-10" | 3'-0" | 13'-4" | 6'-8" | 800 | 2134 |

FIG. 156.—Insulation—Double Bunker.

the room where it is to be used—including all operating, interest and depreciating charges—only \$1 per 24 hours, then the thickness of insulation which it is economical to use in order to reduce the cost of refrigeration must be smaller than in the case where the charge of \$2 or \$3 would be made. The cost of power, of water, and of labor are decided factors that must be considered if the thickness of insulation is to be chosen

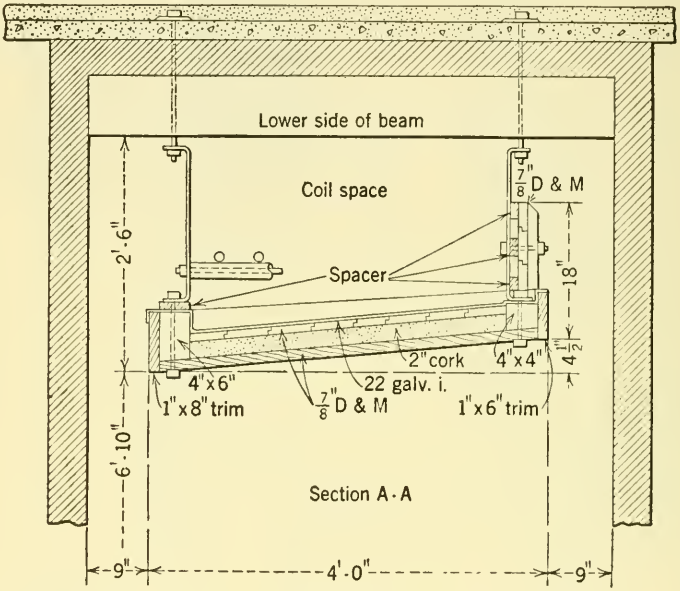



Fig. 157.—Insulation—Bunker Suspended from Ceiling.

which will give the greatest return on the investment. The factors influencing a choice of the economic thickness include then:

1. The cost of a ton of refrigeration delivered, per 24 hours.
2. The cost of the insulation applied, per year.
3. The cost of the refrigerating equipment not included in item 1, and particularly the room piping, etc., per year.
4. The value of the space occupied by the insulation, per year. Considering the case of a wall, ceiling or floor,[5] these losses may be expressed in the following manner:

Let $x$ = the thickness of the insulation, in inches;
$C$ = the coefficient of conduction of the insulation, per 1 in. thick, per hour;

[5] P. Nicholls, The Economic Thickness of Insulation in the Refrigerating Field, Refrigerating Engineering, 1923.

$F$ = the yearly load factor;
$B$ = the cost in dollars for the insulation applied, per 1 in. thick, per 1 sq. ft. of surface;
$A$ = the cost in dollars per ton of refrigeration per 24 hrs., delivered;
$G$ = the cost in dollars per ton of refrigeration of the machinery, etc., not included in $A$;
$I$ = the interest rate as a per cent on the insulation investment;
$R$ = the repair cost per year as a per cent of the insulation first cost;
$Y$ = the life of the insulation, in years;
$I'$, $R'$, $Y'$ = Similar values applied to machinery, etc., in $G$;
$t_a$ = the temperature of the outside air, deg. F., as an average for the period of operations;
$t_m$ = the maximum temperature of the outside air, in deg. F.;
$t$ = the temperature of the cold storage room, in deg. F.;
$t_p$ = the temperature of the refrigerant in the piping, in deg. F.;
$S$ = the value per year of 1 cu. ft. of space in the cold storage room;
$u$ = the coefficient of heat transfer of the wall, per hour, for the materials of construction other than the insulation as given by the usual formula:

$$\frac{1}{u} = \frac{1}{k_1} + \frac{1}{k_2} + \frac{T_1}{C_1} + \frac{T_2}{C_2}, \quad \text{etc.}$$

Using these symbols, the separate costs will be:

1. The cost per year of the heat leakage through the insulation

$$= \frac{t_a - t}{\frac{1}{u} + \frac{x}{C}} \times \frac{24 \times 365 \times F \times A}{288{,}000}$$

2. The cost of the insulation per year, per square foot per 1 in. thick

$$= B \times x\left(\frac{I}{100} + \frac{R}{100} + \frac{1}{Y}\right)$$

3. The cost per year of the investment required to offset the heat leakage through the insulation

$$= \frac{t_m - t}{\frac{1}{u} + \frac{x}{C}} \times \frac{24}{288{,}000} \times G\left(\frac{I'}{100} + \frac{R'}{100} + \frac{1}{Y'}\right)$$

4. The cost of the space occupied by the insulation per year

$$= \frac{Sx}{12}$$

As a rule the cost of insulation, applied, can be expressed by the formula

$$B = \frac{C'}{x} + B' \text{ dollars per board foot,}$$

where

$C'$ is the cost of finish, plaster, nails, labor and overhead per square foot;

and

$B'$ is the cost of the insulation delivered to the job.

Also, if

$P$ is the cost in dollars per square foot of refrigerating piping installed in the cold room as the equipment represented by $G$,

then

$$G = \frac{12{,}000 \times P}{k \times (t - t_p)},$$

where

$k$ = the coefficient of heat transfer for the piping per *hour*.

Then if $Z$ = total cost per year,

$$Z = \frac{t_a - t}{\frac{1}{u} + \frac{x}{C}} \times \frac{365FA}{12{,}000} + \left(\frac{C'}{x} + B'\right) \times \left(\frac{I}{100} + \frac{R}{100} + \frac{1}{Y}\right)$$

$$+ \frac{Sx}{12} + \frac{t_m - t}{\frac{1}{u} + \frac{x}{C}} \times \frac{1}{12{,}000} \times \frac{12{,}000 \times P}{k \times (t - t_p)} \times \left(\frac{I'}{100} + \frac{R'}{100} + \frac{1}{Y'}\right).$$

For a minimum, $dZ/dx = 0$, so, by differentiating and putting this equal to zero and solving for $x$, the result becomes,

$$x = 1.74\sqrt{\frac{A(t_a - t)F + \frac{0.327P}{k(t - t_p)}\left(I' + R' + \frac{100}{Y'}\right)(t_m - t)}{B'\left(I + R + \frac{100}{Y}\right) + 8.3S}} \times C - \frac{C}{u}$$

As an example of the manner in which the value of $x$ may be calculated, the following values may be taken:

The wall may be considered as the equivalent of 12 in. of brick with a $\frac{1}{2}$-in. mortar finish on the cork.

Let

$I$ = 6 per cent;
$I'$ = 6 per cent;
$R$ = 3 per cent;

$R'$ = 3 per cent;
$Y$ = 15 years;
$Y'$ = 8 years;
$k$ = 1.5 B.t.u. per hour;
$C$ = 0.35 B.t.u. per hour;
$t_{average}$ = 50 deg. F.;
$t$ = 0 deg. F.;
$t_{max}$ = 95 deg. F;
$(t - t_p)$ = 10 deg. F.;
$F$ = 1.0;
$A$ = \$2.00;
$S$ = \$0.40 per cu. ft.;
$t_{max} - t$ = 95 deg. F.;
$P$ = \$4.00, the cost of piping per square foot installed, plus all accessories.

$$B = \left(\frac{0.72 + 0.065x}{x}\right);$$

$$G = \frac{288{,}000 \times P}{24 \times k \times (t - t_p)}$$

Placing these values in the equation the value of $x$ becomes 7.0 in. Using a value of \$0.10 for $S$ the value of $x$ is changed to 11.5 in. Solving the general equation for the cost with varying values of $x$, the curve in Fig. 159 is obtained.

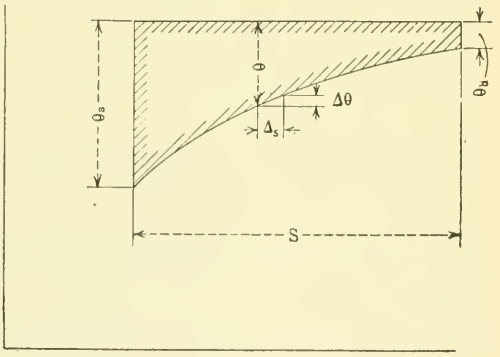



Fig. 158.—Hausbrandt's Formula.

**Heat Transfer; Apparatus.**—More important than the values of heat transfer of building materials are those pertaining to piping, condensers, and coolers of various sorts. The correct design of every portion of the refrigerating plant depends on the proper use of the various heat transfer coefficients. Unfortunately, these values are frequently little better than a guess, in which case the selected values are so liberal as to enable the load imposed on the apparatus to be carried under any set of circumstances, which are very varied in commercial work.

For piping and built-up heat transfer apparatus the total heat

passing through the surface is given by the well-known formula:

$$Q = Ak(t_2 - t_1)$$

where

$A$ = the surface area in square feet;

$k$ = coefficient of heat transfer in B.t.u. per degree difference in temperature per hour per square foot;

$Q$ = B.t.u. per hour;

$(t_2 - t_1)$ = the temperature difference on the two sides of the surface.

Until recently it has been understood by refrigerating engineers that the area of piping is to be taken as the outside area, but the recent test code of the American Society of Mechanical Engineers gives the area as figured from the side of the refrigerant. However, this value as figured is a fictitious value in many cases. The flooded atmospheric and the flooded double-pipe condenser (for example) with at least the lower half of each pipe filled with liquid has its *effective condensing* area reduced in proportion.[6] The double-pipe and the atmospheric condenser with the hot gas entering at the top will certainly lose in effective liquefaction area as con-

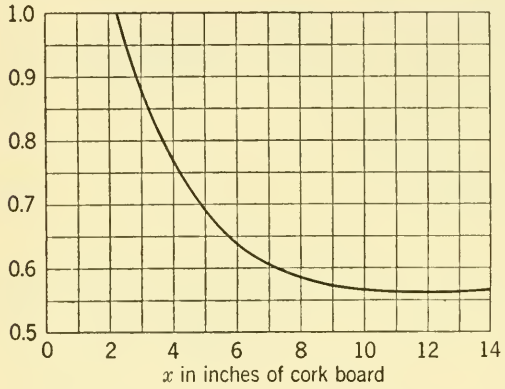



FIG. 159.—Problem—Economic Thickness of Insulation.

[6] The feeling prevails among refrigerating engineers that heat transfer is greater from the metal walls of the condenser to the condensate and then to the gas than from the walls to the saturated refrigerant. In the ammonia condenser the area exposed to the superheated gas is comparatively small in amount. In the bleeder type, for example, this amount is frequently limited to two pipes and only under exceptional circumstances does it increase to three pipes. The remainder of the condenser is in a wet condition with beads of liquid condensate forming on the upper side and running down the inside or dropping to the bottom of the pipe. The refrigerant is *not* exposed to a dry surface but one constantly wet with the liquid. Except in that condenser design which forces the liquid through the pipes at a high velocity—thereby continually breaking down the surface film—there is no advantage, and in fact there is a *positive disadvantage* in permitting the liquid to remain on the surface of the condenser. Once condensed the refrigerant should be drained off as quickly as conditions will permit. In this respect the theoretical design of ammonia condensers follows closely that of the better known and better designed steam surface condensers.

densation occurs and the liquid condensate flows from pipe to pipe into the liquid receiver. The loss of effective surface (loss in the sense of decreased heat transfer due to the layer of liquid ammonia) might be easily 50 per cent or more of the surface in the lower tubes. This trouble is even greater in the case of the drip type condenser, which has a liquid flow in many of the pipes counter to the flow of the compressed gas discharge from the compressor, and always has most of the work of the condenser performed in the upper pipe when cold condensing water is used or in the upper two or three pipes when large amounts of warm water are used.

Another disturbing factor in condensers is the presence of air. In Orrok's[7] tests on steam condensers it was found that the presence of air reduced the value of $k$ to about $\frac{1}{5}$ of the value obtained with an air-free surface.[8] However, all commercial condensers have air present to a greater or a lesser degree, although there is more care now (1927) than

[7] Geo. Orrok, Amer. Soc. of Mech. Eng., 1911.

[8] **The Effect of Surface Film on Heat Transfer.**—Heat transfer in ordinary cases is often confused with the thermal conductivity of the material through which the heat has to pass. For example it is said in the proceedings of an important association that, since copper is seven times as good in conductivity as iron is, replacing iron tubes with copper ones in an evaporator or heater should increase the capacity seven times. Actually it increases the capacity only about 10 per cent.

This indicates that there is some factor in heat transfer for which one must look further. If one considers a plate of copper 1 ft. square and 0.065 in. thick the usual values for the thermal conductivity of copper would indicate that there should flow through the plate, per degree F. difference of temperature on the two sides of the plate per hour, about 40,000 B.t.u. If this were an iron plate, about 6500 B.t.u. should pass through under the same conditions. In actual practice the values taken are usually from 250 to 300 B.t.u. though it is possible that the practical value may go up to as much as 1000 under unusual conditions.

Evidently there is some resistance other than the resistance of the metal itself. The usual explanation is that a *film of almost stagnant liquid* is present on the surface of the metal. Water and most liquids are very poor conductors of heat. It is found, for example, that if a film of water 0.01 in. thick is on one side of a copper plate it will decrease the rate of heat transfer from 40,000 to 300 B.t.u. Evidently then the problem in commercial work is to get heat, not through the metal wall, but through the thin films of stagnant liquids or gas. It is necessary, therefore, to know what factors affect the thickness of these films and how it is possible to predict their resistance.

Consider a tube with water on the inside and steam on the outside, and also the resistance that is offered to the flow of heat from the tube to the liquid. It is found that the greater the velocity of the liquid the thinner the stagnant film and that the greater the viscosity the thicker the film. For a film of given thickness the narrower tubes or denser liquids result in higher rates of heat transfer. These factors can be calculated numerically for a few limited cases which arise in practice. For the great bulk of practical cases it is possible to predict only in part what effect a given change in design may have.

formerly in purging the condenser and in preventing air leakage and air accumulation into the system.

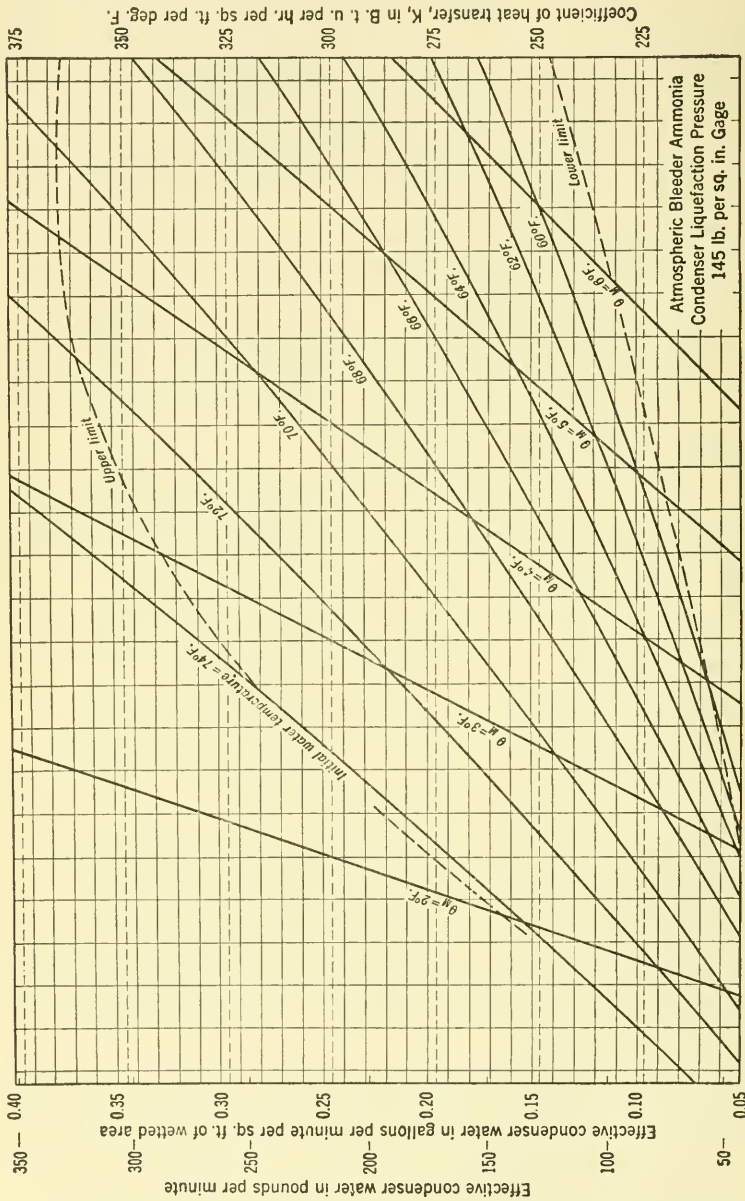



Fig. 160.—Heat Transfer—Atmospheric Type of Bleeder Condenser.

Orrok's tests also brought out that the value of $k$ is dependent also on the mean temperature difference ($t_m$). In tests by the author on a

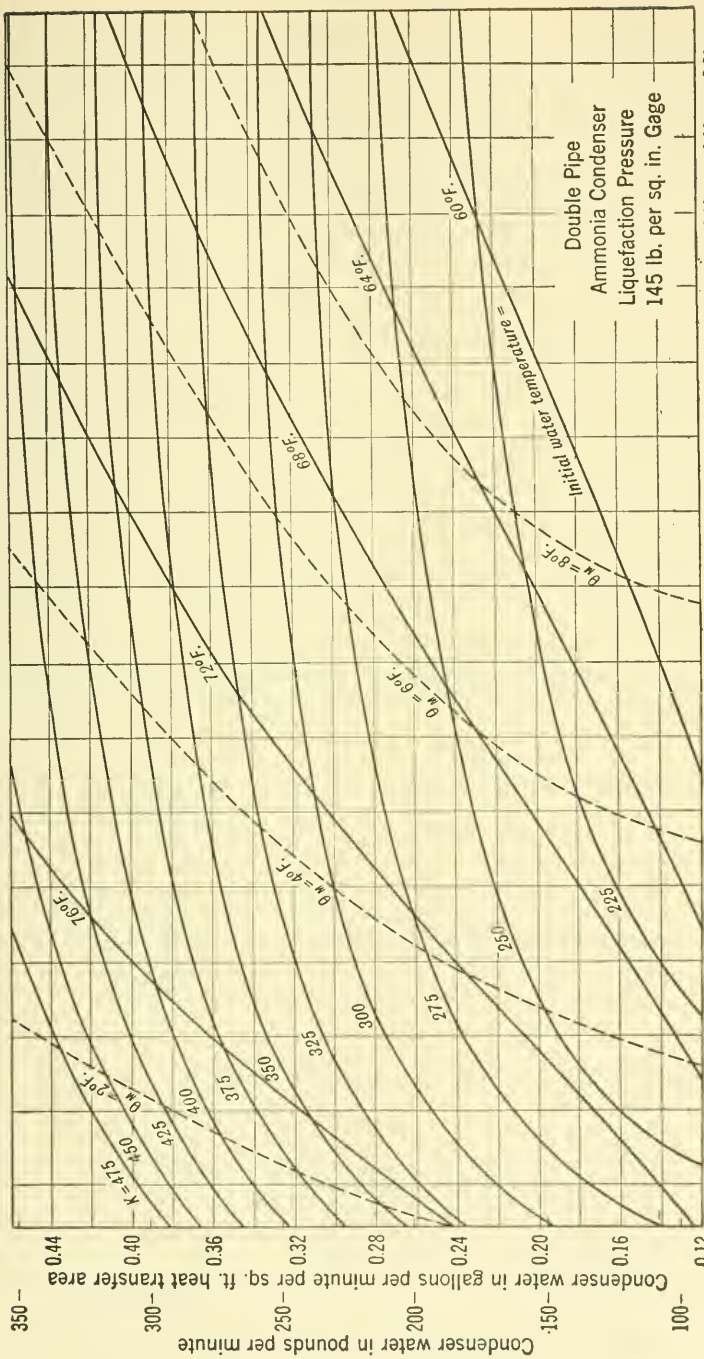



Fig. 161.—Heat Transfer—Double-pipe Ammonia Condenser.

drip type condenser, the value of $k$ was found to increase with a decrease in the mean temperature difference. Showering the condenser with water had the effect of making each pipe more effective, and of decreasing the difference in the amount of liquefaction between the upper and lower pipes, thereby making the lower pipes assume some of the load.[9] In the double pipe and the shell and tube condenser the velocity of the condensing water has a great effect on the value of $k$, but the atmospheric types do not appear to be affected by showering more or less water on them as far as the water flow is concerned.

In regard to the value of $k$ for the cooling of air, water, brine, oil, milk, etc., the cleanliness of the surface, the velocity of the refrigerant and the commodity being cooled—in so far as it affects the surface through which the heat is passing and kind of fluid being cooled—are the factors which affect it. Frostation varies in amount, the piping design permits more or less surface to be non-effective and oils and dirt are permitted to accumulate inside of the evaporator surface. Only the shell and the tube brine cooler has reasonably standard conditions, although the ice plant piping under flooded conditions is uniform in its performance.

The engineer is interested in the working value of $k$, one that can be applied to the formula in the solution of problems, and a few of these are as follows:

| | B.t.u. |
|---|---|
| Can ice making piping: | |
| Old style feed, non-flooded | 12 to 15 |
| Flooded | 20 to 30 |
| Ammonia condensers: | |
| Submerged (obsolete except for $CO_2$) | 30 to 40 |
| Atmospheric, gas entering at top | 60 to 65 |
| Atmospheric, drip or bleeder | 125 to 200 |
| Flooded | 125 to 150 |
| Shell and tube | 150 to 300 |
| Double pipe | 150 to 250 |
| Baudalot coolers, counter flow, atmospheric type: | |
| Milk coolers | 75 |
| Cream coolers | 60 |
| Oil coolers | 10 |
| Water coolers { for direct expansion | 60 |
| Water coolers { for flooded | 80 |
| Brine coolers: | |
| Shell and tube | 90 to 100 |
| Double pipe | 150 to 300 |

[9] The drip pipe condenser was one of 2 in. pipes, 12 high, 20 ft. long.

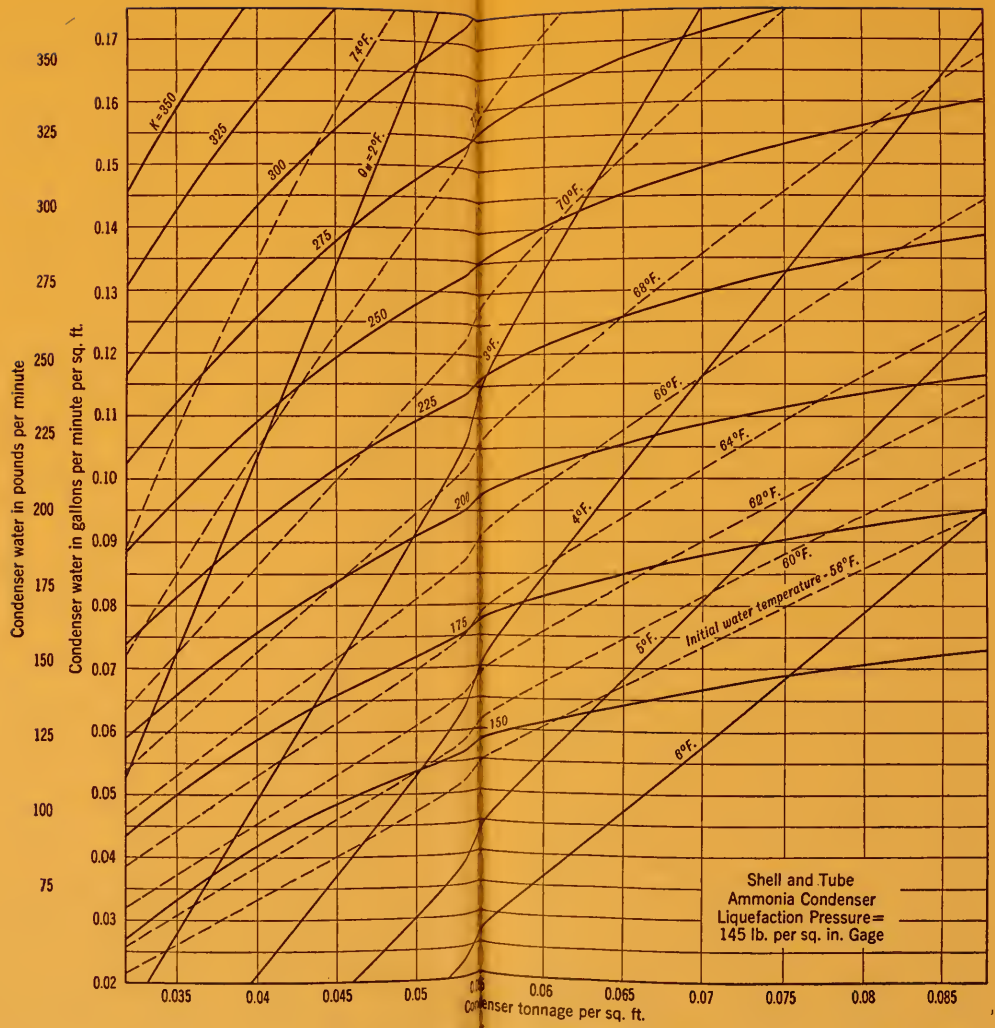



FIG. 162.—Heat Transfer—Shell and Tube Condenser.

*To face page 214.*

| | |
|---|---|
| Cooling coils: | |
| Brine to unagitated air | 2 to 2¼ |
| Direct expansion | 1½ to 2 |
| Water cooler, shell and coil | 15 to 25 |
| Liquid ammonia cooler, shell and coil accumulator | 45 |
| Air dehydrator: | |
| Shell and coil (brine in coil) { 1st coil | 5.0 |
| Shell and coil (brine in coil) { 2d coil | 3.0 |
| Double pipe | 6 to 7 |
| Superheat remover, shell and tube | 15 to 25 |

The action of forced circulation of the air increases $k$ by an amount varying from $1\frac{1}{2}$ to $2\frac{1}{2}$ times the values for unagitated air whereas frost decreases the value of $k$. One inch of frost is considered to decrease the value of $k$ 25 per cent.

**Condenser Surfaces.**—The values usually taken in specifying the size of condensers are as follows, using the value of 260 B.t.u. per ton per minute to be removed by the condensing water. Shell and tube—12 to 15 sq. ft. per ton; shell and coil—16 sq. ft.; double pipe—6 to 10 sq. ft.; and the flooded condenser 5 to 10 sq. ft.

As an example of the method used in dealing with heat transfer the following will be solved:

*Problem.*—A 100-ton ice-making plant (160 tons of refrigeration) makes use of the shell and tube superheat remover. The discharge ammonia gas is cooled to 125 deg. F. Water enters at 70 and is heated to 150 deg. F. Find the amount of water that can be heated (for can dipping or other purposes) and the surface required of the superheat remover, if the condenser pressure is 200 and the suction 35 lb. per sq. in. abs.

The temperature of discharge from the compressor is 234 deg. F. and the amount of heat removed from the ammonia, per pound, is $725 - 655 = 70$ B.t.u.

The number of pounds of ammonia passing through the superheat remover is

$$\frac{200}{613.6 - 150.9} \times 160 \times 60 = 4150.$$

The heat units per hour to be removed $= 70 \times 4150 = 290{,}500$ B.t.u. The amount of water heated is $\frac{290{,}500}{150 - 70} = 3631$ lb. per hour. Taking the value of $k$ as 20.0 per hour, and the logarithmic mean temperature difference as $\frac{84 - 55}{\log_e \frac{84}{55}} = 68.5$ deg. F. Then $290{,}500 = A \times 20 \times 68.5$

$$A = \frac{290{,}500}{1370} = 212 \text{ sq. ft.}$$

The superheat remover should therefore have a heat transfer surface of 212 sq. ft.

**The Mean Temperature Difference.**—In calculating the mean temperature difference the same formula is used where the hotter temperature is constant and the colder is varying (the condenser), the colder constant and the hotter varying (the brine cooler) or the hotter and the

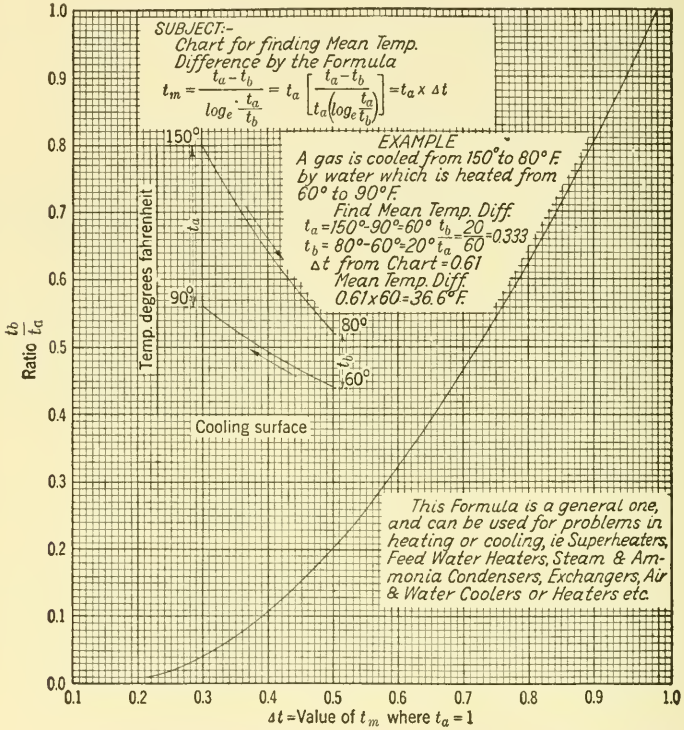



Fig. 163.—Method of Calculating Mean Temperature Difference.

cooler both varying either with parallel or with counter flow of the two fluids. For the first case the formula may be derived as follows:

Referring to Fig. 158, the heat absorbed from a small area $dS$ per unit time is,

$$dQ = CMd\theta$$

where

$C$ = specific heat of the material;

$M$ = weight of the fluid flowing;

$d\theta$ = small change of temperature.

The transfer of heat through the area $dS$ in the same time is

$$dQ = \theta k dS = cM d\theta$$

therefore,

$$\frac{d\theta}{\theta} = \frac{k}{CM} dS,$$

which can be integrated and rearranged to

$$\theta_m = \frac{\theta_a - \theta_b}{\log_e \frac{\theta_a}{\theta_b}}.*$$

Values for this expression are shown in the curve in Fig. 163.

$$* \int_b^a \frac{d\theta}{\theta} = \frac{k}{CM} \int_a^b dS$$

Integrating

$$\log_e \frac{\theta_a}{\theta_b} = \frac{kS}{CM}$$

$$CM(\theta_a - \theta_b) = \frac{kS(\theta_a - \theta_b)}{\log_e \frac{\theta_a}{\theta_b}} = \Delta Q$$

Therefore

$$\frac{\Delta Q}{kS} = \frac{\theta_a - \theta_b}{\log_e \frac{\theta_a}{\theta_b}} = \theta_m$$

**Condenser Tests.**—Figs. 160, 161 and 162 are taken from Bulletin No. 171 of the Engineering Experiment Station of the University of Illinois by Kratz, Macintire and Gould.

TABLE 49

STANDARD BRINE THICKNESS [CORK INSULATION]

| Pipe Size, Inches | Transmission in B.t.u. per Linear Foot per Degree of Difference in Temperature per Hour | Pipe Size, Inches | Transmission in B.t.u. per Linear Foot per Degree of Difference In Temperature per Hour | Pipe Size, Inches | Transmission in B.t.u. per Linear Foot per Degree of Difference in Temperature per Hour |
|---|---|---|---|---|---|
| $\frac{1}{2}$ | 0.141 | 3 | 0.217 | 8 | 0.364 |
| $\frac{3}{4}$ | 0.147 | $3\frac{1}{2}$ | 0.228 | 9 | 0.377 |
| 1 | 0.156 | 4 | 0.259 | 10 | 0.417 |
| $1\frac{1}{4}$ | 0.161 | $4\frac{1}{2}$ | 0.247 | 12 | 0.478 |
| $1\frac{1}{2}$ | 0.165 | 5 | 0.281 | 14 | 0.515 |
| 2 | 0.185 | 6 | 0.293 | 16 | 0.575 |
| $2\frac{1}{2}$ | 0.202 | 7 | 0.354 | | |

SPECIAL THICK BRINE [CORK INSULATION]

| | | | | | |
|---|---|---|---|---|---|
| $\frac{1}{2}$ | 0.122 | 3 | 0.201 | 8 | 0.296 |
| $\frac{3}{4}$ | 0.132 | $3\frac{1}{2}$ | 0.199 | 9 | 0.320 |
| 1 | 0.134 | 4 | 0.221 | 10 | 0.346 |
| $1\frac{1}{4}$ | 0.143 | $4\frac{1}{2}$ | 0.209 | 12 | 0.393 |
| $1\frac{1}{2}$ | 0.153 | 5 | 0.234 | 14 | 0.422 |
| 2 | 0.163 | 6 | 0.249 | 16 | 0.470 |
| $2\frac{1}{2}$ | 0.184 | 7 | 0.273 | | |

ROOMS OR BUILDINGS

| Range of Temperatures, Degrees F. | Thicknesses of Corkboard Recommended for | | | | |
|---|---|---|---|---|---|
| | Walls, inches | Ceilings, inches | Floors on ground, inches | Floors above ground, inches | Roofs, inches |
| Below −15 | 8 | 8 | 7 | 8 | 9 |
| −15 to −5 | 7 | 7 | 6 | 7 | 8 |
| −5 to 10 | 6 | 6 | 5 | 6 | 7 |
| 10 to 25 | 5 | 5 | 4 | 5 | 6 |
| 25 to 40 | 4 | 4 | 3 | 4 | 5 |
| 40 to 50 | 3 | 3 | 2 | 3 | 4 |
| 50 to 65 | 2 | 2 | ..... | 2 | 3 |
| 60 and above | ..... | ..... | ..... | ..... | 2 |

FREEZING TANKS

| | Thicknesses of Corkboard Recommended for | | |
|---|---|---|---|
| | Bottoms, if placed on foundation laid on ground, inches | Temperatures, Degrees F. | Thickness of Corkboard, inches |
| Minimum..... | 5 | −20 to −5 | 8 |
| Preferably..... | 6 | −5 to +5 | 6 |
| | | 5 to 20 | 5 |
| | | 20 to 35 | 4 |
| | | 35 to 45 | 3 |
| | | 45 and above | 2 |

CYLINDRICAL COOLERS, TANKS AND FILTERS FOR COLD LIQUIDS

| Range of Temperatures Degrees F. | Thicknesses of Corkboard Recommended for Sides, Top, Inches |
|---|---|
| Below 0 | 6 |
| 0 to 10 | 5 |
| 10 to 25 | 4 |
| 25 to 45 | 3 |
| 45 to 55 | 2 |
| 55 and above | $1\frac{1}{2}$ |

TABLE 50

[No Void Cork]

| Thickness of Sheets, Inches | Number of Sheets in Crate | Measurement of Sheets | | Size of Crate | | Gross Weight, Pounds |
|---|---|---|---|---|---|---|
| | | Dimensions, inches | Board measure, square feet | Dimensions, inches | Contents cubic feet | |
| 1 | 60 | 36×12×1 | 180 | 38×32×26 | 18.3 | 180 |
| 1½ | 40 | 36×12×1½ | 180 | 38×32×26 | 18.3 | 180 |
| 2 | 28 | 36×12×2 | 168 | 38×30×26 | 17.1 | 168 |
| 3 | 20 | 36×12×3 | 180 | 38×32×26 | 18.3 | 180 |
| 4 | 16 | 36×12×4 | 192 | 38×34×26 | 19.3 | 192 |

TABLE 51

[Armstrong Cork]

Dimensions and Shipping Weights

| Material | Thickness, Inches | Number of Boards per Crate | Square Feet per Crate | Gross Weight per Crate, Pounds | Gross Weight per Square Foot, Pounds | Net Weight per Square Foot, Pounds |
|---|---|---|---|---|---|---|
| Corkboard | 1½ | 12 | 36 | 52 | 1.45 | 1.31 |
| | 2 | 9 | 27 | 51 | 1.89 | 1.70 |
| | 3 | 6 | 18 | 50 | 2.75 | 2.47 |
| | 4 | 4 | 12 | 45 | 3.75 | 3.3 |
| | 6 | 3 | 9 | 45 | 5.00 | 4.48 |

All corkboard is made in sheets 12×36 in. The standard size shipping crate (for 1½-in. to 6-in. thick boards) measures 38×21×13 in., the cubical contents being six cubic feet.

As cork is a natural product and hence does not run uniform in weight, all weights shown in the above table are subject to a maximum variation of 10 per cent, over or under.

TABLE 52

SIZES OF COLD STORAGE DOORS

[Jamieson]

| Size Number | Dimensions inside of Frame or Door "In the Clear" | | | | Wall Opening Required | | | | Cubic Feet, Crated | Approximate Shipping, Weight, Pounds |
|---|---|---|---|---|---|---|---|---|---|---|
| | Width | | Height | | Width | | Height | | | |
| | Ft. | In. | Ft. | In. | Ft. | In. | Ft. | In. | | |
| 00 | 2 | 6 | 2 | .. | 3 | 1 | 2 | 5 | 14 | 150 |
| 0 | 2 | .. | 3 | .. | 2 | 7 | 3 | 5 | 15 | 195 |
| 1 | 2 | .. | 6 | .. | 2 | 7 | 6 | 5 | 25 | 295 |
| 2 | 2 | 6 | 6 | .. | 3 | 1 | 6 | 5 | 30 | 320 |
| 3 | 2 | 6 | 6 | 6 | 3 | 1 | 6 | 11 | 32 | 340 |
| 4 | 3 | .. | 6 | .. | 3 | 7 | 6 | 5 | 33 | 350 |
| 5 | 3 | .. | 6 | 6 | 3 | 7 | 6 | 11 | 35 | 385 |
| 6 | 3 | 6 | 6 | .. | 4 | 1 | 6 | 5 | 37 | 400 |
| 7 | 3 | 6 | 6 | 6 | 4 | 1 | 6 | 11 | 39 | 435 |
| 8 | 3 | .. | 7 | .. | 3 | 7 | 7 | 5 | 38 | 410 |
| 9 | 3 | 6 | 7 | .. | 4 | 1 | 7 | 5 | 42 | 435 |
| 10 | 4 | .. | 6 | .. | 4 | 7 | 6 | 5 | 40 | 435 |
| 11 | 4 | .. | 6 | 6 | 4 | 7 | 6 | 11 | 44 | 450 |
| 13 | 4 | 6 | 6 | 6 | 5 | 1 | 6 | 11 | 48 | 540 |
| 14 | 4 | .. | 7 | .. | 4 | 7 | 7 | 5 | 46 | 530 |
| 15 | 4 | 6 | 7 | .. | 5 | 1 | 7 | 5 | 51 | 560 |
| 16 | 5 | .. | 6 | 6 | 5 | 7 | 6 | 11 | 51 | 585 |
| 17 | 5 | .. | 7 | .. | 5 | 7 | 7 | 5 | 55 | 600 |

SIZES OF TRACK DOORS

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| 25 | 3 | 6 | 6 | $7\frac{1}{2}$ | 4 | 1 | 7 | 10 | 57 | 510 |
| 26 | 3 | 6 | 9 | $7\frac{1}{2}$ | 4 | 1 | 10 | 10 | 79 | 700 |
| 27 | 4 | .. | 6 | $7\frac{1}{2}$ | 4 | 7 | 7 | 10 | 62 | 550 |
| 28 | 4 | .. | 9 | $7\frac{1}{2}$ | 4 | 7 | 10 | 10 | 85 | 780 |

The "Height in the Clear" for the track doors above are suitable for tracks measuring 7 ft. 2 in. and 10 ft. 2 in., respectively, from finished floor level to top of rail.

## CHAPTER VII

## REFRIGERANTS

In Gottsche's Die Kälte-Maschinen (1915) there are 27 German firms listed as builders of ammonia compressors, 29 of carbon dioxide and 17 of sulphide dioxide compressors. In Great Britain—due possibly to law but also to the relatively cold condensing water—the carbonic compressor has been much more popular than that using ammonia up to recent years, whereas the sulphur dioxide type has been used relatively very little. Apparently in all countries the tendency is to use the ammonia system in preference to all others. In the United States ammonia has nearly the entire tonnage except for certain localities and kinds of work where the carbonic system has the preference and for small compressors where sulphur dioxide or some of the special refrigerants are coming into use. Viewed impartially, sulphur dioxide[1] has no present excuse for being used, for the charge of the refrigerant and all oil entering the system has to be *anhydrous* or the acid formed with the water will start corrosion. A leak occurring at a joint, for instance, will also start corrosion sufficient to put the pipe into bad shape in a short time. The only successful sulphur dioxide compressor in the United States in capacities up to 10 tons of refrigeration is the one where the machine is very self-contained and hermetically sealed at the factory.

The matter of detection of leaks is a factor of great importance, especially in the fractional tonnage machines where service has to be supplied, and where the initial charge of the refrigerant is from 3 to 5 lb. This factor is of particular moment in the case of certain refrigerants which, like methyl chloride, do not react chemically with any convenient "indicator" as ammonia and sulphur dioxide do.

The factors influencing the choice of a refrigerant are several in number, as follows: A good refrigerant should be non-corrosive to the piping, the compressor, gages, etc.; it should be harmless to people and commodities; it should be of nominal condenser pressures and yet

[1] According to W. S. Douglas, of Wm. Douglas & Sons, in a letter to Cold Storage, May, 1925, the use of $SO_2$ in Great Britain is increasing steadily, and he considers it the best refrigerant for small machines on account of the low pressure, the lubricating properties, and the ease with which it can be handled.

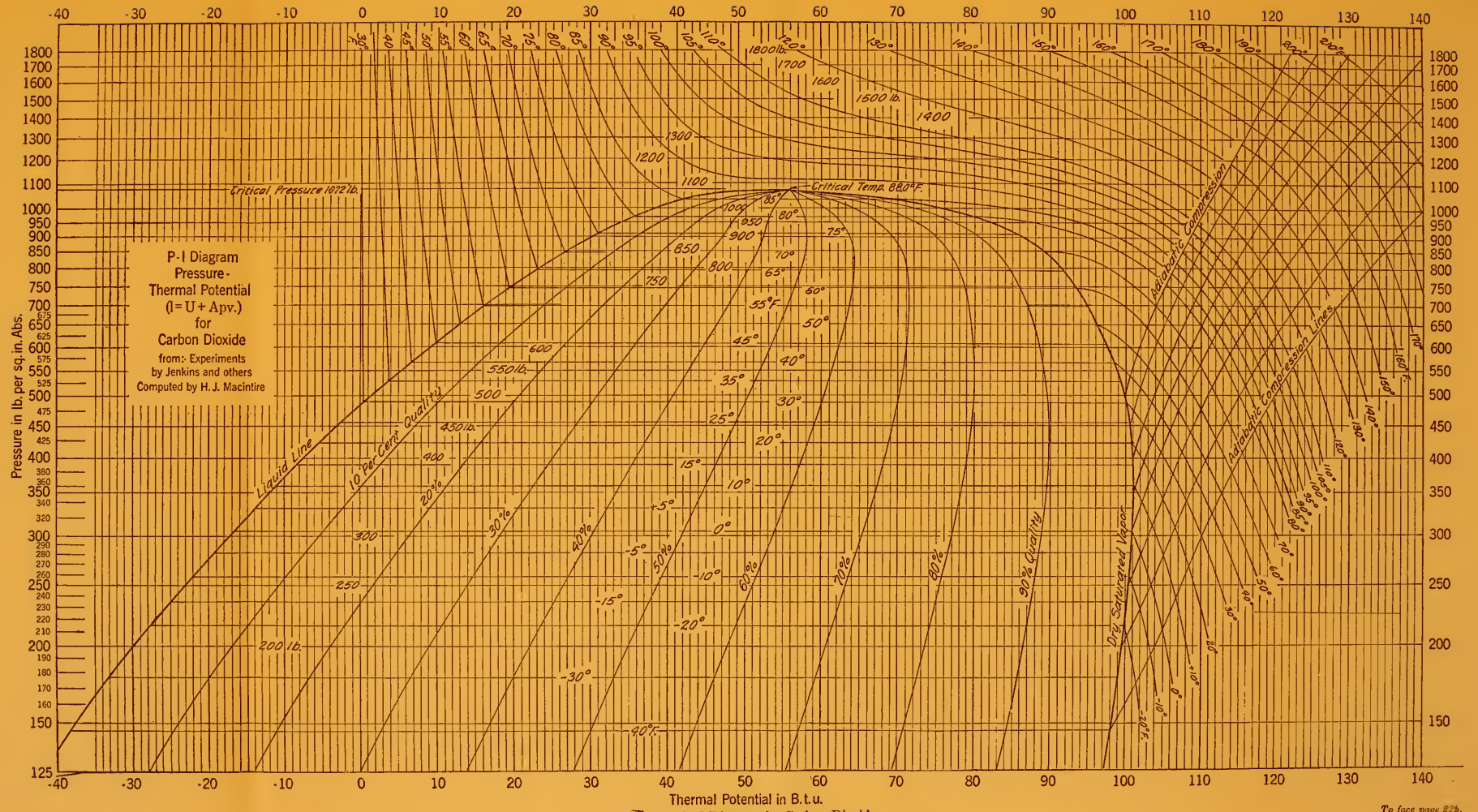



FIG. 164.—P–I Diagram for Carbon Dioxide.

*To face page 224.*

high suction pressures—300 to 450 lb. per sq. in.—permit a low value of the piston displacement per ton of refrigeration, but because of these heavy unit pressures the cylinder diameter is kept as low as possible by the use of a long stroke, the ratio of the stroke to the diameter being from $3\frac{1}{2}$ to 4. Table 53 gives a good idea of the comparison of the refrigerants, using carbon dioxide as unity in the last column. Table 54 gives values for the thermodynamic properties of carbon dioxide.

TABLE 54

THERMODYNAMIC PROPERTIES OF CARBON DIOXIDE

| Temperature, Degrees F. | Pressure, Pounds per Square Inch | Thermal Potential of the Liquid, B.t.u. per Pound | Latent Head of Vaporization | Thermal Potential of Saturated Vapor, B.t.u. per Pound | Density in Cubic Feet per Pound | | Entropy | |
|---|---|---|---|---|---|---|---|---|
| | | | | | Of the liquid | Of the vapor | Of the liquid | Of the vapor |
| −40 | 145.87 | −38.5 | 136.5 | 98.0 | 0.0143 | 0.607 | −0.0850 | 0.2400 |
| −35 | 161.33 | −35.8 | 134.3 | 98.5 | 0.0145 | 0.550 | −0.0793 | 0.2367 |
| −30.0 | 177.97 | −33.1 | 132.1 | 99.0 | 0.0146 | 0.498 | −0.0735 | 0.2336 |
| −25.0 | 195.85 | −30.4 | 129.8 | 99.4 | 0.0148 | 0.451 | −0.0676 | 0.2306 |
| −20.0 | 215.02 | −27.7 | 127.5 | 99.8 | 0.0149 | 0.409 | −0.0619 | 0.2277 |
| −15.0 | 235.53 | −24.9 | 125.0 | 100.1 | 0.0151 | 0.371 | −0.0560 | 0.2250 |
| −10.0 | 257.46 | −22.1 | 122.4 | 100.3 | 0.0153 | 0.337 | −0.0500 | 0.2220 |
| − 5.0 | 280.85 | −19.4 | 120.0 | 100.6 | 0.0155 | 0.307 | −0.0440 | 0.2198 |
| 0.0 | 305.76 | −16.7 | 117.5 | 100.8 | 0.0157 | 0.280 | −0.0381 | 0.2173 |
| 5.0 | 332.2 | −14.0 | 115.0 | 101.0 | 0.0159 | 0.256 | −0.0322 | 0.2151 |
| 10.0 | 360.4 | −11.2 | 112.2 | 101.0 | 0.0161 | 0.235 | −0.0264 | 0.2124 |
| 15.0 | 390.2 | − 8.4 | 109.4 | 101.0 | 0.0164 | 0.216 | −0.0204 | 0.2100 |
| 20.0 | 421.8 | − 5.5 | 106.3 | 100.8 | 0.0166 | 0.199 | −0.0144 | 0.2071 |
| 25.0 | 455.3 | − 2.5 | 103.1 | 100.6 | 0.0169 | 0.182 | −0.0083 | 0.2043 |
| 30.0 | 490.6 | + 0.4 | 99.7 | 100.1 | 0.0172 | 0.168 | −0.0021 | 0.2012 |
| 35.0 | 528.0 | 3.5 | 95.8 | 99.3 | 0.0175 | 0.155 | +0.0039 | 0.1975 |
| 40.0 | 567.3 | 6.6 | 91.8 | 98.4 | 0.0178 | 0.143 | 0.0099 | 0.1934 |
| 45.0 | 608.9 | 9.8 | 87.5 | 97.3 | 0.0182 | 0.132 | 0.0160 | 0.1892 |
| 50.0 | 652.7 | 12.9 | 83.2 | 96.1 | 0.0186 | 0.122 | 0.0220 | 0.1852 |
| 55.0 | 698.8 | 16.1 | 78.7 | 94.8 | 0.0191 | 0.112 | 0.0282 | 0.1809 |
| 60.0 | 747.4 | 19.4 | 74.0 | 93.4 | 0.0197 | 0.093 | 0.0345 | 0.1767 |
| 65.0 | 798.6 | 22.9 | 68.9 | 91.8 | 0.0204 | 0.085 | 0.0412 | 0.1724 |
| 70.0 | 852.5 | 26.6 | 62.7 | 89.3 | 0.0212 | 0.077 | 0.0482 | 0.1665 |
| 75.0 | 909.3 | 30.9 | 54.8 | 85.7 | 0.0222 | 0.070 | 0.0562 | 0.1587 |
| 80.0 | 969.3 | 35.6 | 44.0 | 79.6 | 0.0238 | 0.063 | 0.0649 | 0.1464 |
| 85.0 | 1032.7 | 41.7 | 27.5 | 69.2 | 0.0266 | 0.057 | 0.0761 | 0.1265 |
| 88.0 | 1072.1 | Critical Point | | | | | | |

**Ammonia.**—Ammonia has nominal unit pressures, never being much more than 185 lb. per sq. in., although the presence of air in the condenser may give a higher value on the gage, or extremely high temperature of the condensing water may send the pressure over 200 lb. The value (Table 55) of the latent heat of vaporization ($r$) is high—from 550 to 590 B.t.u.—at usual evaporating temperatures, but the gas is extremely noxious to people and has a bad effect on commodities if heavy leaks occur. As leaks are harder to prevent and are more expensive than they would be for steam under similar pressures, the result is a special form of flanged joint (see Chapter IV). Tests[3] seem to indicate that ammonia is an explosive for certain confined mixtures with air from 13.1 to 26.8 per cent by volume, and some very destructive ammonia explosions have occurred from this cause. Ammonia is found to be corrosive to copper and copper compositions if water is present, and therefore steel and iron are used exclusively in the circuit, and the valves and gages are especially constructed with this point in view. Great care is taken against accidents and the possible loss of life. Where liquid gage glasses are used, special automatic shut-off valves are employed to safeguard against the effect of the glass breaking, and remote control stop valves are frequently used in the larger plants to close the mains near the compressor should a cylinder head or other trouble occur to the compressor.

The decomposition of ammonia has been a subject of interest for years. Lowenstein gives as the result of much testing that no decomposition occurs if the ammonia is pure, even with pressures of 300 lb. for a few days at a time. If water dropped on the piston rod this would work into the cylinder. Certain lubricants used for cylinder lubrication were found to decompose in part and the hydrogen component to collect in the condenser. If so-called foul gases are found in the condenser, and good mineral oil is used, it is very nearly positive that these gases are air. If the gases burn it is the ammonia coming out with the air as it is not practical to purge without the loss of ammonia.

**Sulphur Dioxide.**—Sulphur dioxide has much lower pressures than ammonia but is corrosive to both iron and steel if water is present, even in small amounts. It is very noxious both to people and commodities, although not injurious to *plant life* in small amounts in the air. The latent heat of vaporization ($r$) is about 170 B.t.u. per lb. and the piston displacement per ton of refrigeration is nearly three times that of ammonia for the same operating conditions. It is not a combustible and there is no danger of explosion from leaks or in the event of accidents to the compressor or to the pipe system. In spite of the natural lubricat-

[3] Arthur Lowenstein, American Society of Brewing Technology, March 29, 1916.

## TABLE 55

### Saturated Ammonia: Temperature Table

| Temperature, Degrees F. | Pressure: Absolute, pounds per inch² | Pressure: Gage, pounds per inch² | Volume of Vapor, Feet³ per Pound | Density of Vapor, Pounds per Foot³ | Heat Content: Liquid, B.t.u. per pound | Heat Content: Vapor, B.t.u. per pound | Latent Heat, B.t.u. per Pound | Entropy: Liquid, B.t.u. per pound deg. F. | Entropy: Vapor, B.t.u. per pound deg. F. | Temperature, Degrees F. |
|---|---|---|---|---|---|---|---|---|---|---|
| $t$ | $p$ | $g.p.$ | $V$ | $\frac{1}{V}$ | $h$ | $H$ | $L$ | $s$ | $S$ | $t$ |
| **−60** | 5.55 | 18.6* | 44.73 | 0.02235 | −21.2 | 589.6 | 610.8 | −0.0517 | 1.4769 | **−60** |
| −59 | 5.74 | 18.2* | 43.37 | .02306 | −20.1 | 590.0 | 610.1 | − .0490 | .4741 | −59 |
| −58 | 5.93 | 17.8* | 42.05 | .02378 | −19.1 | 590.4 | 609.5 | − .0464 | .4713 | −58 |
| −57 | 6.13 | 17.4* | 40.79 | .02452 | −18.0 | 590.8 | 608.8 | − .0438 | .4686 | −57 |
| −56 | 6.33 | 17.0* | 39.56 | .02528 | −17.0 | 591.2 | 608.2 | − .0412 | .4658 | −56 |
| **−55** | 6.54 | 16.6* | 38.38 | 0.02605 | −15.9 | 591.6 | 607.5 | −0.0386 | 1.4631 | **−55** |
| −54 | 6.75 | 16.2* | 37.24 | .02685 | −14.8 | 592.1 | 606.9 | − .0360 | .4604 | −54 |
| −53 | 6.97 | 15.7* | 36.15 | .02766 | −13.8 | 592.4 | 606.2 | − .0334 | .4577 | −53 |
| −52 | 7.20 | 15.3* | 35.09 | .02850 | −12.7 | 592.9 | 605.6 | − .0307 | .4551 | −52 |
| −51 | 7.43 | 14.8* | 34.06 | .02936 | −11.7 | 593.2 | 604.9 | − .0281 | .4524 | −51 |
| **−50** | 7.67 | 14.3* | 33.08 | 0.03023 | −10.6 | 593.7 | 604.3 | −0.0256 | 1.4497 | **−50** |
| −49 | 7.91 | 13.8* | 32.12 | .03113 | − 9.6 | 594.0 | 603.6 | − .0230 | .4471 | −49 |
| −48 | 8.16 | 13.3* | 31.20 | .03205 | − 8.5 | 594.4 | 602.9 | − .0204 | .4445 | −48 |
| −47 | 8.42 | 12.8* | 30.31 | .03299 | − 7.4 | 594.9 | 602.3 | − .0179 | .4419 | −47 |
| −46 | 8.68 | 12.2* | 29.45 | .03395 | − 6.4 | 595.2 | 601.6 | − .0153 | .4393 | −46 |
| **−45** | 8.95 | 11.7* | 28.62 | 0.03494 | − 5.3 | 595.6 | 600.9 | −0.0127 | 1.4368 | **−45** |
| −44 | 9.23 | 11.1* | 27.82 | .03595 | − 4.3 | 596.0 | 600.3 | − .0102 | .4342 | −44 |
| −43 | 9.51 | 10.6* | 27.04 | .03698 | − 3.2 | 596.4 | 599.6 | − .0076 | .4317 | −43 |
| −42 | 9.81 | 10.0* | 26.29 | .03804 | − 2.1 | 596.8 | 598.9 | − .0051 | .4292 | −42 |
| −41 | 10.10 | 9.3* | 25.56 | .03912 | − 1.1 | 597.2 | 598.3 | − .0025 | .4267 | −41 |
| **−40** | 10.41 | 8.7* | 24.86 | 0.04022 | 0.0 | 597.6 | 597.6 | 0.0000 | 1.4242 | **−40** |
| −39 | 10.72 | 8.1* | 24.18 | .04135 | 1.1 | 598.0 | 596.9 | .0025 | .4217 | −39 |
| −38 | 11.04 | 7.4* | 23.53 | .04251 | 2.1 | 598.3 | 596.2 | .0051 | .4193 | −38 |
| −37 | 11.37 | 6.8* | 22.89 | .04369 | 3.2 | 598.7 | 595.5 | .0076 | .4169 | −37 |
| −36 | 11.71 | 6.1* | 22.27 | .04489 | 4.3 | 599.1 | 594.8 | .0101 | .4144 | −36 |
| **−35** | 12.05 | 5.4* | 21.68 | 0.04613 | 5.3 | 599.5 | 594.2 | 0.0126 | 1.4120 | **−35** |
| −34 | 12.41 | 4.7* | 21.10 | .04739 | 6.4 | 599.9 | 593.5 | .0151 | .4096 | −34 |
| −33 | 12.77 | 3.9* | 20.54 | .04868 | 7.4 | 600.2 | 592.8 | .0176 | .4072 | −33 |
| −32 | 13.14 | 3.2* | 20.00 | .04999 | 8.5 | 600.6 | 592.1 | .0201 | .4048 | −32 |
| −31 | 13.52 | 2.4* | 19.48 | .05134 | 9.6 | 601.0 | 591.4 | .0226 | .4025 | −31 |
| **−30** | 13.90 | 1.6* | 18.97 | 0.05271 | 10.7 | 601.4 | 590.7 | 0.0250 | 1.4001 | **−30** |
| −29 | 14.30 | 0.8* | 18.48 | .05411 | 11.7 | 601.7 | 590.0 | .0275 | .3978 | −29 |
| −28 | 14.71 | 0.0 | 18.00 | .05555 | 12.8 | 602.1 | 589.3 | .0300 | .3955 | −28 |
| −27 | 15.12 | 0.4 | 17.54 | .05701 | 13.9 | 602.5 | 588.6 | .0325 | .3932 | −27 |
| −26 | 15.55 | 0.8 | 17.09 | .05850 | 14.9 | 602.8 | 587.9 | .0350 | .3909 | −26 |
| **−25** | 15.98 | 1.3 | 16.66 | 0.06003 | 16.0 | 603.2 | 587.2 | 0.0374 | 1.3886 | **−25** |
| −24 | 16.42 | 1.7 | 16.24 | .06158 | 17.1 | 603.6 | 586.5 | .0399 | .3863 | −24 |
| −23 | 16.88 | 2.2 | 15.83 | .06317 | 18.1 | 603.9 | 585.8 | .0423 | .3840 | −23 |
| −22 | 17.34 | 2.6 | 15.43 | .06479 | 19.2 | 604.3 | 585.1 | .0448 | .3818 | −22 |
| −21 | 17.81 | 3.1 | 15.05 | .06644 | 20.3 | 604.6 | 584.3 | .0472 | .3796 | −21 |
| **−20** | 18.30 | 3.6 | 14.68 | 0.06813 | 21.4 | 605.0 | 583.6 | 0.0497 | 1.3774 | **−20** |
| −19 | 18.79 | 4.1 | 14.32 | .06985 | 22.4 | 605.3 | 582.9 | .0521 | .3752 | −19 |
| −18 | 19.30 | 4.6 | 13.97 | .07161 | 23.5 | 605.7 | 582.2 | .0545 | .3729 | −18 |
| −17 | 19.81 | 5.1 | 13.62 | .07340 | 24.6 | 606.1 | 581.5 | .0570 | .3708 | −17 |
| −16 | 20.34 | 5.6 | 13.29 | .07522 | 25.6 | 606.4 | 580.8 | .0594 | .3686 | −16 |
| **−15** | 20.88 | 6.2 | 12.97 | 0.07709 | 26.7 | 606.7 | 580.0 | 0.0618 | 1.3664 | **−15** |
| −14 | 21.43 | 6.7 | 12.66 | .07898 | 27.8 | 607.1 | 579.3 | .0642 | .3643 | −14 |
| −13 | 21.99 | 7.3 | 12.36 | .08092 | 28.9 | 607.5 | 578.6 | .0666 | .3621 | −13 |
| −12 | 22.56 | 7.9 | 12.06 | .08289 | 30.0 | 607.8 | 577.8 | .0690 | .3600 | −12 |
| −11 | 23.15 | 8.5 | 11.78 | .08490 | 31.0 | 608.1 | 577.1 | .0714 | .3579 | −11 |
| **−10** | 23.74 | 9.0 | 11.50 | 0.08695 | 32.1 | 608.5 | 576.4 | 0.0738 | 1.3558 | **−10** |

* Inches of mercury below one standard atmosphere (29.92 in.).

TABLE 55—*Continued*

| Temperature, Degrees F. | Pressure: Absolute, pounds per inch² | Pressure: Gage, pounds per inch² | Volume of Vapor, Feet³ per Pound | Density of Vapor, Pounds per Foot³ | Heat Content: Liquid, B.t.u. per pound | Heat Content: Vapor, B.t.u. per pound | Latent Heat, B.t.u. per Pound | Entropy: Liquid, B.t.u. per pound deg. F. | Entropy: Vapor, B.t.u. per pound deg. F. | Temperature, Degrees F. |
|---|---|---|---|---|---|---|---|---|---|---|
| $t$ | $p$ | $g.p.$ | $V$ | $\frac{1}{V}$ | $h$ | $H$ | $L$ | $s$ | $S$ | $t$ |
| **−10** | 23.74 | 9.0 | 11.50 | 0.08695 | 32.1 | 608.5 | 576.4 | 0.0738 | 1.3558 | **−10** |
| − 9 | 24.35 | 9.7 | 11.23 | .08904 | 33.2 | 608.8 | 575.6 | .0762 | .3537 | − 9 |
| − 8 | 24.97 | 10.3 | 10.97 | .09117 | 34.3 | 609.2 | 574.9 | .0786 | .3516 | − 8 |
| − 7 | 25.61 | 10.9 | 10.71 | .09334 | 35.4 | 609.5 | 574.1 | .0809 | .3495 | − 7 |
| − 6 | 26.26 | 11.6 | 10.47 | .09555 | 36.4 | 609.8 | 573.4 | .0833 | .3474 | − 6 |
| − 5 | 26.92 | 12.2 | 10.23 | 0.09780 | 37.5 | 610.1 | 572.6 | 0.0857 | 1.3454 | **− 5** |
| − 4 | 27.59 | 12.9 | 9.991 | .1001 | 38.6 | 610.5 | 571.9 | .0880 | .3433 | − 4 |
| − 3 | 28.28 | 13.6 | 9.763 | .1024 | 39.7 | 610.8 | 571.1 | .0904 | .3413 | **− 3** |
| − 2 | 28.98 | 14.3 | 9.541 | .1048 | 40.7 | 611.1 | 570.4 | .0928 | .3393 | − 2 |
| − 1 | 29.69 | 15.0 | 9.326 | .1072 | 41.8 | 611.4 | 569.6 | .0951 | .3372 | − 1 |
| **0** | 30.42 | 15.7 | 9.116 | 0.1097 | 42.9 | 611.8 | 568.9 | 0.0975 | 1.3352 | **0** |
| 1 | 31.16 | 16.5 | 8.912 | .1122 | 44.0 | 612.1 | 568.1 | .0998 | .3332 | 1 |
| 2 | 31.92 | 17.2 | 8.714 | .1148 | 45.1 | 612.4 | 567.3 | .1022 | .3312 | 2 |
| 3 | 32.69 | 18.0 | 8.521 | .1174 | 46.2 | 612.7 | 566.5 | .1045 | .3292 | **3** |
| 4 | 33.47 | 18.8 | 8.333 | .1200 | 47.2 | 613.0 | 565.8 | .1069 | .3273 | 4 |
| **5** | 34.27 | 19.6 | 8.150 | 0.1227 | 48.3 | 613.3 | 565.0 | 0.1092 | 1.3253 | **5** |
| 6 | 35.09 | 20.4 | 7.971 | .1254 | 49.4 | 613.6 | 564.2 | .1115 | .3234 | 6 |
| **7** | 35.92 | 21.2 | 7.798 | .1282 | 50.5 | 613.9 | 563.4 | .1138 | .3214 | **7** |
| 8 | 36.77 | 22.1 | 7.629 | .1311 | 51.6 | 614.3 | 562.7 | .1162 | .3195 | 8 |
| 9 | 37.63 | 22.9 | 7.464 | .1340 | 52.7 | 614.6 | 561.9 | .1185 | .3176 | 9 |
| **10** | 38.51 | 23.8 | 7.304 | 0.1369 | 53.8 | 614.9 | 561.1 | 0.1208 | 1.3157 | **10** |
| 11 | 39.40 | 24.7 | 7.148 | .1399 | 54.9 | 615.2 | 560.3 | .1231 | .3137 | 11 |
| 12 | 40.31 | 25.6 | 6.996 | .1429 | 56.0 | 615.5 | 559.5 | .1254 | .3118 | 12 |
| 13 | 41.24 | 26.5 | 6.847 | .1460 | 57.1 | 615.8 | 558.7 | .1277 | .3099 | 13 |
| 14 | 42.18 | 27.5 | 6.703 | .1492 | 58.2 | 616.1 | 557.9 | .1300 | .3081 | 14 |
| **15** | 43.14 | 28.4 | 6.562 | 0.1524 | 59.2 | 616.3 | 557.1 | 0.1323 | 1.3062 | **15** |
| 16 | 44.12 | 29.4 | 6.425 | .1556 | 60.3 | 616.6 | 556.3 | .1346 | .3043 | 16 |
| 17 | 45.12 | 30.4 | 6.291 | .1590 | 61.4 | 616.9 | 555.5 | .1369 | .3025 | 17 |
| 18 | 46.13 | 31.4 | 6.161 | .1623 | 62.5 | 617.2 | 554.7 | .1392 | .3006 | 18 |
| 19 | 47.16 | 32.5 | 6.034 | .1657 | 63.6 | 617.5 | 553.9 | .1415 | .2988 | 19 |
| **20** | 48.21 | 33.5 | 5.910 | 0.1692 | 64.7 | 617.8 | 553.1 | 0.1437 | 1.2969 | **20** |
| 21 | 49.28 | 34.6 | 5.789 | .1728 | 65.8 | 618.0 | 552.2 | .1460 | .2951 | 21 |
| 22 | 50.36 | 35.7 | 5.671 | .1763 | 66.9 | 618.3 | 551.4 | .1483 | .2933 | 22 |
| 23 | 51.47 | 36.8 | 5.556 | .1800 | 68.0 | 618.6 | 550.6 | .1505 | .2915 | 23 |
| 24 | 52.59 | 37.9 | 5.443 | .1837 | 69.1 | 618.9 | 549.8 | .1528 | .2897 | 24 |
| **25** | 53.73 | 39.0 | 5.334 | 0.1875 | 70.2 | 619.1 | 548.9 | 0.1551 | 1.2879 | **25** |
| 26 | 54.90 | 40.2 | 5.227 | .1913 | 71.3 | 619.4 | 548.1 | .1573 | .2861 | 26 |
| 27 | 56.08 | 41.4 | 5.123 | .1952 | 72.4 | 619.7 | 547.3 | .1596 | .2843 | 27 |
| 28 | 57.28 | 42.6 | 5.021 | .1992 | 73.5 | 619.9 | 546.4 | .1618 | .2825 | 28 |
| 29 | 58.50 | 43.8 | 4.922 | .2032 | 74.6 | 620.2 | 545.6 | .1641 | .2808 | 29 |
| **30** | 59.74 | 45.0 | 4.825 | 0.2073 | 75.7 | 620.5 | 544.8 | 0.1663 | 1.2790 | **30** |
| 31 | 61.00 | 46.3 | 4.730 | .2114 | 76.8 | 620.7 | 543.9 | .1686 | .2773 | 31 |
| 32 | 62.29 | 47.6 | 4.637 | .2156 | 77.9 | 621.0 | 543.1 | .1708 | .2755 | 32 |
| 33 | 63.59 | 48.9 | 4.547 | .2199 | 79.0 | 621.2 | 542.2 | .1730 | .2738 | 33 |
| 34 | 64.91 | 50.2 | 4.459 | .2243 | 80.1 | 621.5 | 541.4 | .1753 | .2721 | 34 |
| **35** | 66.26 | 51.6 | 4.373 | 0.2287 | 81.2 | 621.7 | 540.5 | 0.1775 | 1.2704 | **35** |
| 36 | 67.63 | 52.9 | 4.289 | .2332 | 82.3 | 622.0 | 539.7 | .1797 | .2686 | 36 |
| 37 | 69.02 | 54.3 | 4.207 | .2377 | 83.4 | 622.2 | 538.8 | .1819 | .2669 | 37 |
| 38 | 70.43 | 55.7 | 4.126 | .2423 | 84.6 | 622.5 | 537.9 | .1841 | .2652 | 38 |
| 39 | 71.87 | 57.2 | 4.048 | .2470 | 85.7 | 622.7 | 537.0 | .1863 | .2635 | 39 |
| **40** | 73.32 | 58.6 | 3.971 | 0.2518 | 86.8 | 623.0 | 536.2 | 0 .1885 | 1.2618 | **40** |

TABLE 55—*Continued*

| Temperature, Degrees F. | Pressure | | Volume of Vapor, Feet³ per Pound | Density of Vapor, Pounds per Foot³ | Heat Content | | Latent Heat, B.t.u. per Pound | Entropy | | Temperature, Degrees F. |
|---|---|---|---|---|---|---|---|---|---|---|
| | Absolute, pounds per inch² | Gage, pounds per inch² | | | Liquid, B.t.u. per pound | Vapor, B.t.u. per pound | | Liquid, B.t.u. per pound deg. F. | Vapor, B.t.u. per pound deg. F. | |
| | | | | $\frac{1}{V}$ | | | | | | |
| $t$ | $p$ | $g.p.$ | $V$ | | $h$ | $H$ | $L$ | $s$ | $S$ | $t$ |
| **40** | 73.32 | 58.6 | 3.971 | 0.2518 | 86.8 | 623.0 | 536.2 | 0.1885 | 1.2618 | **40** |
| 41 | 74.80 | 60.1 | 3.897 | .2566 | 87.9 | 623.2 | 535.3 | .1908 | .2602 | 41 |
| 42 | 76.31 | 61.6 | 3.823 | .2616 | 89.0 | 623.4 | 534.4 | .1930 | .2585 | 42 |
| 43 | 77.83 | 63.1 | 3.752 | .2665 | 90.1 | 623.7 | 533.6 | .1952 | .2568 | 43 |
| 44 | 79.38 | 64.7 | 3.682 | .2716 | 91.2 | 623.9 | 532.7 | .1974 | .2552 | 44 |
| **45** | 80.96 | 66.3 | 3.614 | 0.2767 | 92.3 | 624.1 | 531.8 | 0.1996 | 1.2535 | **45** |
| 46 | 82.55 | 67.9 | 3.547 | .2819 | 93.5 | 624.4 | 530.9 | .2018 | .2519 | 46 |
| 47 | 84.18 | 69.5 | 3.481 | .2872 | 94.6 | 624.6 | 530.0 | .2040 | .2502 | 47 |
| 48 | 85.82 | 71.1 | 3.418 | .2926 | 95.7 | 624.8 | 529.1 | .2062 | .2486 | 48 |
| 49 | 87.49 | 72.8 | 3.355 | .2981 | 96.8 | 625.0 | 528.2 | .2083 | .2469 | 49 |
| **50** | 89.19 | 74.5 | 3.294 | 0.3036 | 97.9 | 625.2 | 527.3 | 0.2105 | 1.2453 | **50** |
| 51 | 90.91 | 76.2 | 3.234 | .3092 | 99.1 | 625.5 | 526.4 | .2127 | .2437 | 51 |
| 52 | 92.66 | 78.0 | 3.176 | .3149 | 100.2 | 625.7 | 525.5 | .2149 | .2421 | 52 |
| 53 | 94.43 | 79.7 | 3.119 | .3207 | 101.3 | 625.9 | 524.6 | .2171 | .2405 | 53 |
| 54 | 96.23 | 81.5 | 3.063 | .3265 | 102.4 | 626.1 | 523.7 | .2192 | .2389 | 54 |
| **55** | 98.06 | 83.4 | 3.008 | 0.3325 | 103.5 | 626.3 | 522.8 | 0.2214 | 1.2373 | **55** |
| 56 | 99.91 | 85.2 | 2.954 | .3385 | 104.7 | 626.5 | 521.8 | .2236 | .2357 | 56 |
| 57 | 101.8 | 87.1 | 2.902 | .3446 | 105.8 | 626.7 | 520.9 | .2257 | .2341 | 57 |
| 58 | 103.7 | 89.0 | 2.851 | .3508 | 106.9 | 626.9 | 520.0 | .2279 | .2325 | 58 |
| 59 | 105.6 | 90.9 | 2.800 | .3571 | 108.1 | 627.1 | 519.0 | .2301 | .2310 | 59 |
| **60** | 107.6 | 92.9 | 2.751 | 0.3635 | 109.2 | 627.3 | 518.1 | 0.2322 | 1.2294 | **60** |
| 61 | 109.6 | 94.9 | 2.703 | .3700 | 110.3 | 627.5 | 517.2 | .2344 | .2278 | 61 |
| 62 | 111.6 | 96.9 | 2.656 | .3765 | 111.5 | 627.7 | 516.2 | .2365 | .2262 | 62 |
| 63 | 113.6 | 98.9 | 2.610 | .3832 | 112.6 | 627.9 | 515.3 | .2387 | .2247 | 63 |
| 64 | 115.7 | 101.0 | 2.565 | .3899 | 113.7 | 628.0 | 514.3 | .2408 | .2231 | 64 |
| **65** | 117.8 | 103.1 | 2.520 | 0.3968 | 114.8 | 628.2 | 513.4 | 0.2430 | 1.2216 | **65** |
| 66 | 120.0 | 105.3 | 2.477 | .4037 | 116.0 | 628.4 | 512.4 | .2451 | .2201 | 66 |
| 67 | 122.1 | 107.4 | 2.435 | .4108 | 117.1 | 628.6 | 511.5 | .2473 | .2186 | 67 |
| 68 | 124.3 | 109.6 | 2.393 | .4179 | 118.3 | 628.8 | 510.5 | .2494 | .2170 | 68 |
| 69 | 126.5 | 111.8 | 2.352 | .4251 | 119.4 | 628.9 | 509.5 | .2515 | .2155 | 69 |
| **70** | 128.8 | 114.1 | 2.312 | 0.4325 | 120.5 | 629.1 | 508.6 | 0.2537 | 1.2140 | **70** |
| 71 | 131.1 | 116.4 | 2.273 | .4399 | 121.7 | 629.3 | 507.6 | .2558 | .2125 | 71 |
| 72 | 133.4 | 118.7 | 2.235 | .4474 | 122.8 | 629.4 | 506.6 | .2579 | .2110 | 72 |
| 73 | 135.7 | 121.0 | 2.197 | .4551 | 124.0 | 629.6 | 505.6 | .2601 | .2095 | 73 |
| 74 | 138.1 | 123.4 | 2.161 | .4628 | 125.1 | 629.8 | 504.7 | .2622 | .2080 | 74 |
| **75** | 140.5 | 125.8 | 2.125 | 0.4707 | 126.2 | 629.9 | 503.7 | 0.2643 | 1.2065 | **75** |
| 76 | 143.0 | 128.3 | 2.089 | .4786 | 127.4 | 630.1 | 502.7 | .2664 | .2050 | 76 |
| 77 | 145.4 | 130.7 | 2.055 | .4867 | 128.5 | 630.2 | 501.7 | .2685 | .2035 | 77 |
| 78 | 147.9 | 133.2 | 2.021 | .4949 | 129.7 | 630.4 | 500.7 | .2706 | .2020 | 78 |
| 79 | 150.5 | 135.8 | 1.988 | .5031 | 130.8 | 630.5 | 499.7 | .2728 | .2006 | 79 |
| **80** | 153.0 | 138.3 | 1.955 | 0.5115 | 132.0 | 630.7 | 498.7 | 0.2749 | 1.1991 | **80** |
| 81 | 155.6 | 140.9 | 1.923 | .5200 | 133.1 | 630.8 | 497.7 | .2769 | .1976 | 81 |
| 82 | 158.3 | 143.6 | 1.892 | .5287 | 134.3 | 631.0 | 496.7 | .2791 | .1962 | 82 |
| 83 | 161.0 | 146.3 | 1.861 | .5374 | 135.4 | 631.1 | 495.7 | .2812 | .1947 | 83 |
| 84 | 163.7 | 149.0 | 1.831 | .5462 | 136.6 | 631.3 | 494.7 | .2833 | .1933 | 84 |
| **85** | 166.4 | 151.7 | 1.801 | 0.5552 | 137.8 | 631.4 | 493.6 | 0.2854 | 1.1918 | **85** |

TABLE 55—*Continued*

| Temperature, Degrees F. | Pressure | | Volume of Vapor, Feet³ per Pound | Density of Vapor, Pounds per Foot³ | Heat Content | | Latent Heat, B.t.u. per Pound | Entropy | | Tem- at De |
|---|---|---|---|---|---|---|---|---|---|---|
| | Absolute, pounds per inch² | Gage, pounds per inch² | | | Liquid, B.t.u. per pound | Vapor, B.t.u. per pound | | Liquid, B.t.u. per pound deg. F. | Vapor, B.t.u. per pound deg. F. | |
| $t$ | $p$ | $g.p.$ | $V$ | $\frac{1}{V}$ | $h$ | $H$ | $L$ | $s$ | $S$ | |
| **85** | 166.4 | 151.7 | 1.801 | 0.5552 | 137.8 | 631.4 | 493.6 | 0.2854 | 1.1918 | |
| 86 | 169.2 | 154.5 | 1.772 | .5643 | 138.9 | 631.5 | 492.6 | .2875 | .1904 | |
| 87 | 172.0 | 157.3 | 1.744 | .5735 | 140.1 | 631.7 | 491.6 | .2895 | .1889 | |
| 88 | 174.8 | 160.1 | 1.716 | .5828 | 141.2 | 631.8 | 490.6 | .2917 | .1875 | |
| 89 | 177.7 | 163.0 | 1.688 | .5923 | 142.4 | 631.9 | 489.5 | .2937 | .1860 | |
| **90** | 180.6 | 165.9 | 1.661 | 0.6019 | 143.5 | 632.0 | 488.5 | 0.2958 | 1.1846 | |
| 91 | 183.6 | 168.9 | 1.635 | .6116 | 144.7 | 632.1 | 487.4 | .2979 | .1832 | |
| 92 | 186.6 | 171.9 | 1.609 | .6214 | 145.8 | 632.2 | 486.4 | .3000 | .1818 | |
| 93 | 189.6 | 174.9 | 1.584 | .6314 | 147.0 | 632.3 | 485.3 | .3021 | .1804 | |
| 94 | 192.7 | 178.0 | 1.559 | .6415 | 148.2 | 632.5 | 484.3 | .3041 | .1789 | |
| **95** | 195.8 | 181.1 | 1.534 | 0.6517 | 149.4 | 632.6 | 483.2 | 0.3062 | 1.1775 | |
| 96 | 198.9 | 184.2 | 1.510 | .6620 | 150.5 | 632.6 | 482.1 | .3083 | .1761 | |
| 97 | 202.1 | 187.4 | 1.487 | .6725 | 151.7 | 632.8 | 481.1 | .3104 | .1747 | |
| 98 | 205.3 | 190.6 | 1.464 | .6832 | 152.9 | 632.9 | 480.0 | .3125 | .1733 | |
| 99 | 208.6 | 193.9 | 1.441 | .6939 | 154.0 | 632.9 | 478.9 | .3145 | .1719 | |
| **100** | 211.9 | 197.2 | 1.419 | 0.7048 | 155.2 | 633.0 | 477.8 | 0.3166 | 1.1705 | 1 |
| 101 | 215.2 | 200.5 | 1.397 | .7159 | 156.4 | 633.1 | 476.7 | .3187 | .1691 | 1 |
| 102 | 218.6 | 203.9 | 1.375 | .7270 | 157.6 | 633.2 | 475.6 | .3207 | .1677 | 1 |
| 103 | 222.0 | 207.3 | 1.354 | .7384 | 158.7 | 633.3 | 474.6 | .3228 | .1663 | 1 |
| 104 | 225.4 | 210.7 | 1.334 | .7498 | 159.9 | 633.4 | 473.5 | .3248 | .1649 | 1 |
| **105** | 228.9 | 214.2 | 1.313 | 0.7615 | 161.1 | 633.4 | 472.3 | 0.3269 | 1.1635 | 1 |
| 106 | 232.5 | 217.8 | 1.293 | .7732 | 162.3 | 633.5 | 471.2 | .3289 | .1621 | 1 |
| 107 | 236.0 | 221.3 | 1.274 | .7852 | 163.5 | 633.6 | 470.1 | .3310 | .1607 | 1 |
| 108 | 239.7 | 225.0 | 1.254 | .7972 | 164.6 | 633.6 | 469.0 | .3330 | .1593 | 1 |
| 109 | 243.3 | 228.6 | 1.235 | .8095 | 165.8 | 633.7 | 467.9 | .3351 | .1580 | 1 |
| **110** | 247.0 | 232.3 | 1.217 | 0.8219 | 167.0 | 633.7 | 466.7 | 0.3372 | 1.1566 | 1 |
| 111 | 250.8 | 236.1 | 1.198 | .8344 | 168.2 | 633.8 | 465.6 | .3392 | .1552 | 1 |
| 112 | 254.5 | 239.8 | 1.180 | .8471 | 169.4 | 633.8 | 464.4 | .3413 | .1538 | 1 |
| 113 | 258.4 | 243.7 | 1.163 | .8600 | 170.6 | 633.9 | 463.3 | .3433 | .1524 | 1 |
| 114 | 262.2 | 247.5 | 1.145 | .8730 | 171.8 | 633.9 | 462.1 | .3453 | .1510 | 1 |
| **115** | 266.2 | 251.5 | 1.128 | 0.8862 | 173.0 | 633.9 | 460.9 | 0.3474 | 1.1497 | 1 |
| 116 | 270.1 | 255.4 | 1.112 | .8996 | 174.2 | 634.0 | 459.8 | .3495 | .1483 | 1 |
| 117 | 274.1 | 259.4 | 1.095 | .9132 | 175.4 | 634.0 | 458.6 | .3515 | .1469 | 1 |
| 118 | 278.2 | 263.5 | 1.079 | .9269 | 176.6 | 634.0 | 457.4 | .3535 | .1455 | 1 |
| 119 | 282.3 | 267.6 | 1.063 | .9408 | 177.8 | 634.0 | 456.2 | .3556 | .1441 | 1 |
| **120** | 286.4 | 271.7 | 1.047 | 0.9549 | 179.0 | 634.0 | 455.0 | 0.3576 | 1.1427 | 1 |
| 121 | 290.6 | 275.9 | 1.032 | .9692 | 180.2 | 634.0 | 453.8 | .3597 | .1414 | 1 |
| 122 | 294.8 | 280.1 | 1.017 | .9837 | 181.4 | 634.0 | 452.6 | .3618 | .1400 | 1 |
| 123 | 299.1 | 284.4 | 1.002 | .9983 | 182.6 | 634.0 | 451.4 | .3638 | .1386 | 1 |
| 124 | 303.4 | 288.7 | 0.987 | 1.0132 | 183.9 | 634.0 | 450.1 | .3659 | .1372 | 1 |
| **125** | 307.8 | 293.1 | 0.973 | 1.028 | 185.1 | 634.0 | 448.9 | 0.3679 | 1.1358 | 1 |

## TABLE 55*a*

### Saturated Ammonia: Absolute Pressure Table

| Pressure (Absolute), Pounds per Inch² | Temperature, Degrees F. | Volume of Vapor, Feet³ per Pound | Density of Vapor, Pounds per Foot³ | Heat Content | | Latent Heat, B.t.u. per Pound | Entropy | | | Pressure (Absolute), Pounds per Inch² |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Liquid, B.t.u. per pound | Vapor, B.t.u per pound | | Liquid, B.t.u. per pound deg. F. | Evap., B.t.u. per pound deg. F. | Vapor, B.t.u. per pound deg. F. | |
| | | | $\frac{1}{V}$ | | | | | $\frac{L}{T}$ | | |
| $p$ | $t$ | $V$ | | $h$ | $H$ | $L$ | $s$ | | $S$ | $p$ |
| 5.0 | −63.11 | 49.31 | 0.02029 | −24.5 | 588.3 | 612.8 | −0.0599 | 1.5456 | 1.4857 | 5.0 |
| 5.5 | −60.27 | 45.11 | .02217 | −21.5 | 589.5 | 611.0 | − .0524 | .5301 | .4777 | 5.5 |
| 6.0 | −57.64 | 41.59 | .02405 | −18.7 | 590.6 | 609.3 | − .0455 | .5158 | .4703 | 6.0 |
| 6.5 | −55.18 | 38.59 | .02591 | −16.1 | 591.6 | 607.7 | − .0390 | .5026 | .4636 | 6.5 |
| 7.0 | −52.88 | 36.01 | .02777 | −13.7 | 592.5 | 606.2 | − .0330 | .4904 | .4574 | 7.0 |
| 7.5 | −50.70 | 33.77 | 0.02962 | −11.3 | 593.4 | 604.7 | −0.0274 | 1.4790 | 1.4516 | 7.5 |
| 8.0 | −48.64 | 31.79 | .03146 | − 9.2 | 594.2 | 603.4 | − .0221 | .4683 | .4462 | 8.0 |
| 8.5 | −46.69 | 30.04 | .03329 | − 7.1 | 595.0 | 602.1 | − .0171 | .4582 | .4411 | 8.5 |
| 9.0 | −44.83 | 28.48 | .03511 | − 5.1 | 595.7 | 600.8 | − .0123 | .4486 | .4363 | 9.0 |
| 9.5 | −43.05 | 27.08 | .03693 | − 3.2 | 596.4 | 599.6 | − .0077 | .4396 | .4319 | 9.5 |
| 10.0 | −41.34 | 25.81 | 0.03874 | − 1.4 | 597.1 | 598.5 | −0.0034 | 1.4310 | 1.4276 | 10.0 |
| 10.5 | −39.71 | 24.66 | .04055 | + 0.3 | 597.7 | 597.4 | + .0007 | .4228 | .4235 | 10.5 |
| 11.0 | −38.14 | 23.61 | .04235 | 2.0 | 598.3 | 596.3 | .0047 | .4149 | .4196 | 11.0 |
| 11.5 | −36.62 | 22.65 | .04414 | 3.6 | 598.9 | 595.3 | .0085 | .4074 | .4159 | 11.5 |
| 12.0 | −35.16 | 21.77 | .04593 | 5.1 | 599.4 | 594.3 | .0122 | .4002 | .4124 | 12.0 |
| 12.5 | −33.74 | 20.96 | 0.04772 | 6.7 | 600.0 | 593.3 | 0.0157 | 1.3933 | 1.4090 | 12.5 |
| 13.0 | −32.37 | 20.20 | .04950 | 8.1 | 600.5 | 592.4 | .0191 | .3866 | .4057 | 13.0 |
| 13.5 | −31.05 | 19.50 | .05128 | 9.6 | 601.0 | 591.4 | .0225 | .3801 | .4026 | 13.5 |
| 14.0 | −29.76 | 18.85 | .05305 | 10.9 | 601.4 | 590.5 | .0257 | .3739 | .3996 | 14.0 |
| 14.5 | −28.51 | 18.24 | .05482 | 12.2 | 601.9 | 589.7 | .0288 | .3679 | .3967 | 14.5 |
| 15.0 | −27.29 | 17.67 | 0.05658 | 13.6 | 602.4 | 588.8 | 0.0318 | 1.3620 | 1.3938 | 15.0 |
| 15.5 | −26.11 | 17.14 | .05834 | 14.8 | 602.8 | 588.0 | .0347 | .3564 | .3911 | 15.5 |
| 16.0 | −24.95 | 16.64 | .06010 | 16.0 | 603.2 | 587.2 | .0375 | .3510 | .3885 | 16.0 |
| 16.5 | −23.83 | 16.17 | .06186 | 17.2 | 603.6 | 586.4 | .0403 | .3456 | .3859 | 16.5 |
| 17.0 | −22.73 | 15.72 | .06361 | 18.4 | 604.0 | 585.6 | .0430 | .3405 | .3835 | 17.0 |
| 17.5 | −21.66 | 15.30 | 0.06535 | 19.6 | 604.4 | 584.8 | 0.0456 | 1.3354 | 1.3810 | 17.5 |
| 18.0 | −20.61 | 14.90 | .06710 | 20.7 | 604.8 | 584.1 | .0482 | .3305 | .3787 | 18.0 |
| 18.5 | −19.59 | 14.53 | .06884 | 21.8 | 605.1 | 583.3 | .0507 | .3258 | .3765 | 18.5 |
| 19.0 | −18.58 | 14.17 | .07058 | 22.9 | 605.5 | 582.6 | .0531 | .3211 | .3742 | 19.0 |
| 19.5 | −17.60 | 13.83 | .07232 | 23.9 | 605.8 | 581.9 | .0555 | .3166 | .3721 | 19.5 |
| 20.0 | −16.64 | 13.50 | 0.07405 | 25.0 | 606.2 | 581.2 | 0.0578 | 1.3122 | 1.3700 | 20.0 |
| 20.5 | −15.70 | 13.20 | .07578 | 26.0 | 606.5 | 580.5 | .0601 | .3078 | .3679 | 20.5 |
| 21.0 | −14.78 | 12.90 | .07751 | 27.0 | 606.8 | 579.8 | .0623 | .3036 | .3659 | 21.0 |
| 21.5 | −13.87 | 12.62 | .07924 | 27.9 | 607.1 | 579.2 | .0645 | .2995 | .3640 | 21.5 |
| 22.0 | −12.98 | 12.35 | .08096 | 28.9 | 607.4 | 578.5 | .0666 | .2955 | .3621 | 22.0 |
| 22.5 | −12.11 | 12.09 | 0.08268 | 29.8 | 607.7 | 577.9 | 0.0687 | 1.2915 | 1.3602 | 22.5 |
| 23.0 | −11.25 | 11.85 | .08440 | 30.8 | 608.1 | 577.3 | .0708 | .2876 | .3584 | 23.0 |
| 23.5 | −10.41 | 11.61 | .08612 | 31.7 | 608.3 | 576.6 | .0728 | .2838 | .3566 | 23.5 |
| 24.0 | − 9.58 | 11.39 | .08783 | 32.6 | 608.6 | 576.0 | .0748 | .2801 | .3549 | 24.0 |
| 24.5 | − 8.76 | 11.17 | .08955 | 33.5 | 608.9 | 575.4 | .0768 | .2764 | .3532 | 24.5 |
| 25.0 | − 7.96 | 10.96 | 0.09126 | 34.3 | 609.1 | 574.8 | 0.0787 | 1.2728 | 1.3515 | 25.0 |
| 25.5 | − 7.17 | 10.76 | .09297 | 35.2 | 609.4 | 574.2 | .0805 | .2693 | .3498 | 25.5 |
| 26.0 | − 6.39 | 10.56 | .09468 | 36.0 | 609.7 | 573.7 | .0824 | .2658 | .3482 | 26.0 |
| 26.5 | − 5.63 | 10.38 | .09638 | 36.8 | 609.9 | 573.1 | .0842 | .2625 | .3467 | 26.5 |
| 27.0 | − 4.87 | 10.20 | .09809 | 37.7 | 610.2 | 572.5 | .0860 | .2591 | .3451 | 27.0 |
| 27.5 | − 4.13 | 10.02 | 0.09979 | 38.4 | 610.4 | 572.0 | 0.0878 | 1.2558 | 1.3436 | 27.5 |
| 28.0 | − 3.40 | 9.853 | .1015 | 39.3 | 610.7 | 571.4 | .0895 | .2526 | .3421 | 28.0 |
| 28.5 | − 2.68 | 9.691 | .1032 | 40.0 | 610.9 | 570.9 | .0912 | .2494 | .3406 | 28.5 |
| 29.0 | − 1.97 | 9.534 | .1049 | 40.8 | 611.1 | 570.3 | .0929 | .2463 | .3392 | 29.0 |
| 29.5 | − 1.27 | 9.383 | .1066 | 41.6 | 611.4 | 569.8 | .0945 | .2433 | .3378 | 29.5 |
| 30.0 | − 0.57 | 9.236 | 0.1083 | 42.3 | 611.6 | 569.3 | 0.0962 | 1.2402 | 1.3364 | 30.0 |

TABLE 55a—*Continued*

| Pressure (Absolute) Pounds per Inch² | Temperature, Degrees F. | Volume of Vapor, Feet³ per Pound | Density of Vapor, Pounds per Foot³ | Heat Content | | Latent Heat, B.t.u. per Pound | Entropy | | | Pressure (Absolute), Pounds per Inch² |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Liquid, B.t.u. per pound | Vapor, B.t.u. per pound | | Liquid, B.t.u. per pound deg. F. | Evap., B.t.u. per pound deg. F. | Vapor, B.t.u. per pound deg. F. | |
| | | | $\frac{1}{V}$ | | | | | $\frac{L}{T}$ | | |
| $p$ | $t$ | $V$ | | $h$ | $H$ | $L$ | $s$ | | $S$ | $p$ |
| **30** | −0.57 | 9.236 | 0.1083 | 42.3 | 611.6 | 569.3 | 0.0962 | 1.2402 | 1.3364 | **30** |
| 31 | +0.79 | 8.955 | .1117 | 43.8 | 612.0 | 568.2 | .0993 | .2343 | .3336 | 31 |
| 32 | 2.11 | 8.693 | .1150 | 45.2 | 612.4 | 567.2 | .1024 | .2286 | .3310 | 32 |
| 33 | 3.40 | 8.445 | .1184 | 46.6 | 612.8 | 566.2 | .1055 | .2230 | .3285 | 33 |
| 34 | 4.66 | 8.211 | .1218 | 48.0 | 613.2 | 565.2 | .1084 | .2176 | .3260 | 34 |
| **35** | 5.89 | 7.991 | 0.1251 | 49.3 | 613.6 | 564.3 | 0.1113 | 1.2123 | 1.3236 | **35** |
| 36 | 7.09 | 7.782 | .1285 | 50.6 | 614.0 | 563.4 | .1141 | .2072 | .3213 | 36 |
| 37 | 8.27 | 7.584 | .1319 | 51.9 | 614.3 | 562.4 | .1168 | .2022 | .3190 | 37 |
| 38 | 9.42 | 7.396 | .1352 | 53.2 | 614.7 | 561.5 | .1195 | .1973 | .3168 | 38 |
| 39 | 10.55 | 7.217 | .1386 | 54.4 | 615.0 | 560.6 | .1221 | .1925 | .3146 | 39 |
| **40** | 11.66 | 7.047 | 0.1419 | 55.6 | 615.4 | 559.8 | 0.1246 | 1.1879 | 1.3125 | **40** |
| 41 | 12.74 | 6.885 | .1452 | 56.8 | 615.7 | 558.9 | .1271 | .1833 | .3104 | 41 |
| 42 | 13.81 | 6.731 | .1486 | 57.9 | 616.0 | 558.1 | .1296 | .1788 | .3084 | 42 |
| 43 | 14.85 | 6.583 | .1519 | 59.1 | 616.3 | 557.2 | .1320 | .1745 | .3065 | 43 |
| 44 | 15.88 | 6.442 | .1552 | 60.2 | 616.6 | 556.4 | .1343 | .1703 | .3046 | 44 |
| **45** | 16.88 | 6.307 | 0.1586 | 61.3 | 616.9 | 555.6 | 0.1366 | 1.1661 | 1.3027 | **45** |
| 46 | 17.87 | 6.177 | .1619 | 62.4 | 617.2 | 554.8 | .1389 | .1620 | .3009 | 46 |
| 47 | 18.84 | 6.053 | .1652 | 63.4 | 617.4 | 554.0 | .1411 | .1580 | .2991 | 47 |
| 48 | 19.80 | 5.934 | .1685 | 64.5 | 617.7 | 553.2 | .1433 | .1540 | .2973 | 48 |
| 49 | 20.74 | 5.820 | .1718 | 65.5 | 618.0 | 552.5 | .1454 | .1502 | .2956 | 49 |
| **50** | 21.67 | 5.710 | 0.1751 | 66.5 | 618.2 | 551.7 | 0.1475 | 1.1464 | 1.2939 | **50** |
| 51 | 22.58 | 5.604 | .1785 | 67.5 | 618.5 | 551.0 | .1496 | .1427 | .2923 | 51 |
| 52 | 23.48 | 5.502 | .1818 | 68.5 | 618.7 | 550.2 | .1516 | .1390 | .2906 | 52 |
| 53 | 24.36 | 5.404 | .1851 | 69.5 | 619.0 | 549.5 | .1536 | .1354 | .2890 | 53 |
| 54 | 25.23 | 5.309 | .1884 | 70.4 | 619.2 | 548.8 | .1556 | .1319 | .2875 | 54 |
| **55** | 26.09 | 5.218 | 0.1917 | 71.4 | 619.4 | 548.0 | 0.1575 | 1.1284 | 1.2859 | **55** |
| 56 | 26.94 | 5.129 | .1950 | 72.3 | 619.7 | 547.4 | .1594 | .1250 | .2844 | 56 |
| 57 | 27.77 | 5.044 | .1983 | 73.3 | 619.9 | 546.6 | .1613 | .1217 | .2830 | 57 |
| 58 | 28.59 | 4.962 | .2015 | 74.2 | 620.1 | 545.9 | .1631 | .1184 | .2815 | 58 |
| 59 | 29.41 | 4.882 | .2048 | 75.0 | 620.3 | 545.3 | .1650 | .1151 | .2801 | 59 |
| **60** | 30.21 | 4.805 | 0.2081 | 75.9 | 620.5 | 544.6 | 0.1668 | 1.1119 | 1.2787 | **60** |
| 61 | 31.00 | 4.730 | .2114 | 76.8 | 620.7 | 543.9 | .1685 | .1088 | .2773 | 61 |
| 62 | 31.78 | 4.658 | .2147 | 77.7 | 620.9 | 543.2 | .1703 | .1056 | .2759 | 62 |
| 63 | 32.55 | 4.588 | .2180 | 78.5 | 621.1 | 542.6 | .1720 | .1026 | .2746 | 63 |
| 64 | 33.31 | 4.519 | .2213 | 79.4 | 621.3 | 541.9 | .1737 | .0996 | .2733 | 64 |
| **65** | 34.06 | 4.453 | 0.2245 | 80.2 | 621.5 | 541.3 | 0.1754 | 1.0966 | 1.2720 | **65** |
| 66 | 34.81 | 4.389 | .2278 | 81.0 | 621.7 | 540.7 | .1770 | .0937 | .2707 | 66 |
| 67 | 35.54 | 4.327 | .2311 | 81.8 | 621.9 | 540.1 | .1787 | .0907 | .2694 | 67 |
| 78 | 36.27 | 4.267 | .2344 | 82.6 | 622.0 | 539.4 | .1803 | .0879 | .2682 | 68 |
| 69 | 36.99 | 4.208 | .2377 | 83.4 | 622.2 | 538.8 | .1819 | .0851 | .2670 | 69 |
| **70** | 37.70 | 4.151 | 0.2409 | 84.2 | 622.4 | 538.2 | 0.1835 | 1.0823 | 1.2658 | **70** |
| 71 | 38.40 | 4.095 | .2442 | 85.0 | 622.6 | 537.6 | .1850 | .0795 | .2645 | 71 |
| 72 | 39.09 | 4.041 | .2475 | 85.8 | 622.8 | 537.0 | .1866 | .0768 | .2634 | 72 |
| 73 | 39.78 | 3.988 | .2507 | 86.5 | 622.9 | 536.4 | .1881 | .0741 | .2622 | 73 |
| 74 | 40.46 | 3.937 | .2540 | 87.3 | 623.1 | 535.8 | .1896 | .0715 | .2611 | 74 |
| **75** | 41.13 | 3.887 | 0.2573 | 88.0 | 623.2 | 535.2 | 0.1910 | 1.0689 | 1.2599 | **75** |
| 76 | 41.80 | 3.838 | .2606 | 88.8 | 623.4 | 534.6 | .1925 | .0663 | .2588 | 76 |
| 77 | 42.46 | 3.790 | .2638 | 89.5 | 623.5 | 534.0 | .1940 | .0637 | .2577 | 77 |
| 78 | 43.11 | 3.744 | .2671 | 90.2 | 623.7 | 533.5 | .1954 | .0612 | .2566 | 78 |
| 79 | 43.76 | 3.699 | .2704 | 90.9 | 623.8 | 532.9 | .1968 | .0587 | .2555 | 79 |
| **80** | 44.40 | 3.655 | 0.2736 | 91.7 | 624.0 | 532.3 | 0.1982 | 1.0563 | 1.2545 | **80** |

TABLE 55a—*Continued*

| Pressure (Absolute) Pounds per Inch² $p$ | Temperature, Degrees F. $t$ | Volume of Vapor, Feet³ per Pound $V$ | Density of Vapor, Pounds per Foot³ $\frac{1}{V}$ | Heat Content: Liquid, B.t.u. per pound $h$ | Heat Content: Vapor, B.t.u. per pound $H$ | Latent Heat, B.t.u. per Pound $L$ | Entropy: Liquid, B.t.u. per pound deg. F. $s$ | Entropy: Evap., B.t.u. per pound deg. F. $\frac{L}{T}$ | Entropy: Vapor, B.t.u. per pound deg. F. $S$ | Pressure (Absolute), Pounds per Inch² $p$ |
|---|---|---|---|---|---|---|---|---|---|---|
| **80** | 44.40 | 3.655 | 0.2736 | 91.7 | 624.0 | 532.3 | 0.1982 | 1.0563 | 1.2545 | **80** |
| 81 | 45.03 | 3.612 | .2769 | 92.4 | 624.1 | 531.7 | .1996 | .0538 | .2534 | 81 |
| 82 | 45.66 | 3.570 | .2801 | 93.1 | 624.3 | 531.2 | .2010 | .0514 | .2524 | 82 |
| 83 | 46.28 | 3.528 | .2834 | 93.8 | 624.4 | 530.6 | .2024 | .0490 | .2514 | 83 |
| 84 | 46.89 | 3.488 | .2867 | 94.5 | 624.6 | 530.1 | .2037 | .0467 | .2504 | 84 |
| **85** | 47.50 | 3.449 | 0.2899 | 95.1 | 624.7 | 529.6 | 0.2051 | 1.0443 | 1.2494 | **85** |
| 86 | 48.11 | 3.411 | .2932 | 95.8 | 624.8 | 529.0 | .2064 | .0420 | .2484 | 86 |
| 87 | 48.71 | 3.373 | .2964 | 96.5 | 625.0 | 528.5 | .2077 | .0397 | .2474 | 87 |
| 88 | 49.30 | 3.337 | .2997 | 97.2 | 625.1 | 527.9 | .2090 | .0375 | .2465 | 88 |
| 89 | 49.89 | 3.301 | .3030 | 97.8 | 625.2 | 527.4 | .2103 | .0352 | .2455 | 89 |
| **90** | 50.47 | 3.266 | 0.3062 | 98.4 | 625.3 | 526.9 | 0.2115 | 1.0330 | 1.2445 | **90** |
| 91 | 51.05 | 3.231 | .3095 | 99.1 | 625.5 | 526.4 | .2128 | .0308 | .2436 | 91 |
| 92 | 51.62 | 3.198 | .3127 | 99.8 | 625.6 | 525.8 | .2141 | .0286 | .2427 | 92 |
| 93 | 52.19 | 3.165 | .3160 | 100.4 | 625.7 | 525.3 | .2153 | .0265 | .2418 | 93 |
| 94 | 52.76 | 3.132 | .3192 | 101.0 | 625.8 | 524.8 | .2165 | .0243 | .2408 | 94 |
| **95** | 53.32 | 3.101 | 0.3225 | 101.6 | 625.9 | 524.3 | 0.2177 | 1.0222 | 1.2399 | **95** |
| 96 | 53.87 | 3.070 | .3258 | 102.3 | 626.1 | 523.8 | .2190 | .0201 | .2391 | 96 |
| 97 | 54.42 | 3.039 | .3290 | 102.9 | 626.2 | 523.3 | .2201 | .0181 | .2382 | 97 |
| 98 | 54.97 | 3.010 | .3323 | 103.5 | 626.3 | 522.8 | .2213 | .0160 | .2373 | 98 |
| 99 | 55.51 | 2.980 | .3355 | 104.1 | 626.4 | 522.3 | .2225 | .0140 | .2365 | 99 |
| **100** | 56.05 | 2.952 | 0.3388 | 104.7 | 626.5 | 521.8 | 0.2237 | 1.0119 | 1.2356 | **100** |
| 102 | 57.11 | 2.896 | .3453 | 105.9 | 626.7 | 520.8 | .2260 | .0079 | .2339 | 102 |
| 104 | 58.16 | 2.843 | .3518 | 107.1 | 626.9 | 519.8 | .2282 | .0041 | .2323 | 104 |
| 106 | 59.19 | 2.791 | .3583 | 108.3 | 627.1 | 518.8 | .2305 | 1.0002 | .2307 | 106 |
| 108 | 60.21 | 2.741 | .3648 | 109.4 | 627.3 | 517.9 | .2327 | 0.9964 | .2291 | 108 |
| **110** | 61.21 | 2.693 | 0.3713 | 110.5 | 627.5 | 517.0 | 0.2348 | 0.9927 | 1.2275 | **110** |
| 112 | 62.20 | 2.647 | .3778 | 111.7 | 627.7 | 516.0 | .2369 | .9890 | .2259 | 112 |
| 114 | 63.17 | 2.602 | .3843 | 112.8 | 627.9 | 515.1 | .2390 | .9854 | .2244 | 114 |
| 116 | 64.13 | 2.559 | .3909 | 113.9 | 628.1 | 514.2 | .2411 | .9819 | .2230 | 116 |
| 118 | 65.08 | 2.517 | .3974 | 114.9 | 628.2 | 513.3 | .2431 | .9784 | .2215 | 118 |
| **120** | 66.02 | 2.476 | 0.4039 | 116.0 | 628.4 | 512.4 | 0.2452 | 0.9749 | 1.2201 | **120** |
| 122 | 66.94 | 2.437 | .4104 | 117.1 | 628.6 | 511.5 | .2471 | .9715 | .2186 | 122 |
| 124 | 67.86 | 2.399 | .4169 | 118.1 | 628.7 | 510.6 | .2491 | .9682 | .2173 | 124 |
| 126 | 68.76 | 2.362 | .4234 | 119.1 | 628.9 | 509.8 | .2510 | .9649 | .2159 | 126 |
| 128 | 69.65 | 2.326 | .4299 | 120.1 | 629.0 | 508.9 | .2529 | .9616 | .2145 | 128 |
| **130** | 70.53 | 2.291 | 0.4364 | 121.1 | 629.2 | 508.1 | 0.2548 | 0.9584 | 1.2132 | **130** |
| 132 | 71.40 | 2.258 | .4429 | 122.1 | 629.3 | 507.2 | .2567 | .9552 | .2119 | 132 |
| 134 | 72.26 | 2.225 | .4494 | 123.1 | 629.5 | 506.4 | .2585 | .9521 | .2106 | 134 |
| 136 | 73.11 | 2.193 | .4559 | 124.1 | 629.6 | 505.5 | .2603 | .9490 | .2093 | 136 |
| 138 | 73.95 | 2.162 | .4624 | 125.1 | 629.8 | 504.7 | .2621 | .9460 | .2081 | 138 |
| **140** | 74.79 | 2.132 | 0.4690 | 126.0 | 629.9 | 503.9 | 0.2638 | 0.9430 | 1.2068 | **140** |
| 142 | 75.61 | 2.103 | .4755 | 126.9 | 630.0 | 503.1 | .2656 | .9400 | .2056 | 142 |
| 144 | 76.42 | 2.075 | .4820 | 127.9 | 630.2 | 502.3 | .2673 | .9371 | .2044 | 144 |
| 146 | 77.23 | 2.047 | .4885 | 128.8 | 630.3 | 501.5 | .2690 | .9342 | .2032 | 146 |
| 148 | 78.03 | 2.020 | .4951 | 129.7 | 630.4 | 500.7 | .2707 | .9313 | .2020 | 148 |
| **150** | 78.81 | 1.994 | 0.5016 | 130.6 | 630.5 | 499.9 | 0.2724 | 0.9285 | 1.2009 | **150** |

TABLE 55*a*—*Continued*

| Pressure (Absolute) Pounds per Inch² | Temperature, Degrees F. | Volume of Vapor, Feet³ per Pound | Density of Vapor, Pounds per Foot³ | Heat Content | | Latent Heat, B.t.u. per Pound | Entropy | | | Pressure (Absolute), Pounds per Inch² |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Liquid, B.t.u. per pound | Vapor, B.t.u. per pound | | Liquid, B.t.u. per pound deg. F. | Evap., B.t.u. per pound deg. F. | Vapor, B.t.u. per pound deg. F. | |
| $p$ | $t$ | $V$ | $\frac{1}{V}$ | $h$ | $H$ | $L$ | $s$ | $\frac{L}{T}$ | $S$ | $p$ |
| **150** | 78.81 | 1.994 | 0.5016 | 130.6 | 630.5 | 499.9 | 0.2724 | 0.9285 | 1.2009 | **150** |
| 152 | 79.60 | 1.968 | .5081 | 131.5 | 630.6 | 499.1 | .2740 | .9257 | .1997 | 152 |
| 154 | 80.37 | 1.943 | .5147 | 132.4 | 630.7 | 498.3 | .2756 | .9229 | .1985 | 154 |
| 156 | 81.13 | 1.919 | .5212 | 133.3 | 630.9 | 497.6 | .2772 | .9202 | .1974 | 156 |
| 158 | 81.89 | 1.895 | .5277 | 134.2 | 631.0 | 496.8 | .2788 | .9175 | .1963 | 158 |
| **160** | 82.64 | 1.872 | 0.5343 | 135.0 | 631.1 | 496.1 | 0.2804 | 0.9148 | 1.1952 | **160** |
| 162 | 83.39 | 1.849 | .5408 | 135.9 | 631.2 | 495.3 | .2820 | .9122 | .1942 | 162 |
| 164 | 84.12 | 1.827 | .5473 | 136.8 | 631.3 | 494.5 | .2835 | .9096 | .1931 | 164 |
| 166 | 84.85 | 1.805 | .5539 | 137.6 | 631.4 | 493.8 | .2850 | .9070 | .1920 | 166 |
| 168 | 85.57 | 1.784 | .5604 | 138.4 | 631.5 | 493.1 | .2866 | .9044 | .1910 | 168 |
| **170** | 86.29 | 1.764 | 0.5670 | 139.3 | 631.6 | 492.3 | 0.2881 | 0.9019 | 1.1900 | **170** |
| 172 | 87.00 | 1.744 | .5735 | 140.1 | 631.7 | 491.6 | .2895 | .8994 | .1889 | 172 |
| 174 | 87.71 | 1.724 | .5801 | 140.9 | 631.7 | 490.8 | .2910 | .8969 | .1879 | 174 |
| 176 | 88.40 | 1.705 | .5866 | 141.7 | 631.8 | 490.1 | .2925 | .8944 | .1869 | 176 |
| 178 | 89.10 | 1.686 | .5932 | 142.5 | 631.9 | 489.4 | .2939 | .8920 | .1859 | 178 |
| **180** | 89.78 | 1.667 | 0.5998 | 143.3 | 632.0 | 488.7 | 0.2954 | 0.8896 | 1.1850 | **180** |
| 182 | 90.46 | 1.649 | .6063 | 144.1 | 632.1 | 488.0 | .2968 | .8872 | .1840 | 182 |
| 184 | 91.14 | 1.632 | .6129 | 144.8 | 632.1 | 487.3 | .2982 | .8848 | .1830 | 184 |
| 186 | 91.80 | 1.614 | .6195 | 145.6 | 632.2 | 486.6 | .2996 | .8825 | .1821 | 186 |
| 188 | 92.47 | 1.597 | .6261 | 146.4 | 632.3 | 485.9 | .3010 | .8801 | .1811 | 188 |
| **190** | 93.13 | 1.581 | 0.6326 | 147.2 | 632.4 | 485.2 | 0.3024 | 0.8778 | 1.1802 | **190** |
| 192 | 93.78 | 1.564 | .6392 | 147.9 | 632.4 | 484.5 | .3037 | .8755 | .1792 | 192 |
| 194 | 94.43 | 1.548 | .6458 | 148.7 | 632.5 | 483.8 | .3050 | .8733 | .1783 | 194 |
| 196 | 95.07 | 1.533 | .6524 | 149.5 | 632.6 | 483.1 | .3064 | .8710 | .1774 | 196 |
| 198 | 95.71 | 1.517 | .6590 | 150.2 | 632.6 | 482.4 | .3077 | .8688 | .1765 | 198 |
| **200** | 96.34 | 1.502 | 0.6656 | 150.9 | 632.7 | 481.8 | 0.3090 | 0.8666 | 1.1756 | **200** |
| 205 | 97.90 | 1.466 | .6821 | 152.7 | 632.8 | 480.1 | .3122 | .8612 | .1734 | 205 |
| 210 | 99.43 | 1.431 | .6986 | 154.6 | 633.0 | 478.4 | .3154 | .8559 | .1713 | 210 |
| 215 | 100.94 | 1.398 | .7152 | 156.3 | 633.1 | 476.8 | .3185 | .8507 | .1692 | 215 |
| 220 | 102.42 | 1.367 | .7318 | 158.0 | 633.2 | 475.2 | .3216 | .8455 | .1671 | 220 |
| **225** | 103.87 | 1.336 | 0.7484 | 159.7 | 633.3 | 473.6 | 0.3246 | 0.8405 | 1.1651 | **225** |
| 230 | 105.30 | 1.307 | .7650 | 161.4 | 633.4 | 472.0 | .3275 | .8356 | .1631 | 230 |
| 235 | 106.71 | 1.279 | .7817 | 163.1 | 633.5 | 470.4 | .3304 | .8307 | .1611 | 235 |
| 240 | 108.09 | 1.253 | .7984 | 164.7 | 633.6 | 468.9 | .3332 | .8260 | .1592 | 240 |
| 245 | 109.46 | 1.227 | .8151 | 166.4 | 633.7 | 467.3 | .3360 | .8213 | .1573 | 245 |
| **250** | 110.80 | 1.202 | 0.8319 | 168.0 | 633.8 | 465.8 | 0.3388 | 0.8167 | 1.1555 | **250** |
| 255 | 112.12 | 1.178 | .8487 | 169.5 | 633.8 | 464.3 | .3415 | .8121 | .1536 | 255 |
| 260 | 113.42 | 1.155 | .8655 | 171.1 | 633.9 | 462.8 | .3441 | .8077 | .1518 | 260 |
| 265 | 114.71 | 1.133 | .8824 | 172.6 | 633.9 | 461.3 | .3468 | .8033 | .1501 | 265 |
| 270 | 115.97 | 1.112 | .8993 | 174.1 | 633.9 | 459.8 | .3494 | .7989 | .1483 | 270 |
| **275** | 117.22 | 1.091 | 0.9162 | 175.6 | 634.0 | 458.4 | 0.3519 | 0.7947 | 1.1466 | **275** |
| 280 | 118.45 | 1.072 | .9332 | 177.1 | 634.0 | 456.9 | .3545 | .7904 | .1449 | 280 |
| 285 | 119.66 | 1.052 | .9502 | 178.6 | 634.0 | 455.4 | .3569 | .7863 | .1432 | 285 |
| 290 | 120.86 | 1.034 | .9672 | 180.0 | 634.0 | 454.0 | .3594 | .7821 | .1415 | 290 |
| 295 | 122.05 | 1.016 | .9843 | 181.5 | 634.0 | 452.5 | .3618 | .7781 | .1399 | 295 |
| **300** | 123.21 | 0.999 | 1.0015 | 182.9 | 634.0 | 451.1 | 0.3642 | 0.7741 | 1.1383 | **300** |

## TABLE 55*b*

### Properties of Superheated Ammonia

Pressures Are Pounds per Square Inches Absolute

| Temperature, Degrees F. | 10 — *41.34 Degrees* | | | Temperature, Degrees F. | 15 — *27.29 Degrees* | | | Temperature, Degrees F. | 20 — *16.64 Degrees* | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | *V* | *H* | *S* | | *V* | *H* | *S* | | *V* | *H* | *S* |
| *Saturation* | *25.81* | *597.1* | *1.4276* | *Saturation* | *17.67* | *602.4* | *1.3938* | *Saturation* | *13.50* | *606.2* | *1.3700* |
| −30 | 26.58 | 603.2 | 1.4420 | −20 | 18.01 | 606.4 | 1.4031 | −10 | 13.74 | 610.0 | 1.3784 |
| −20 | 27.26 | 608.5 | .4542 | −10 | 18.47 | 611.9 | .4154 | | | | |
| −10 | 27.92 | 613.7 | .4659 | | | | | **0** | 14.09 | 615.5 | 1.3907 |
| | | | | **0** | 18.92 | 617.2 | 1.4272 | 10 | 14.44 | 621.0 | .4025 |
| **0** | 28.58 | 618.9 | 1.4773 | 10 | 19.37 | 622.5 | .4386 | 20 | 14.78 | 626.4 | .4138 |
| 10 | 29.24 | 624.0 | .4884 | 20 | 19.82 | 627.8 | .4497 | 30 | 15.11 | 631.7 | .4248 |
| 20 | 29.90 | 629.1 | .4992 | 30 | 20.26 | 633.0 | .4604 | 40 | 15.45 | 637.0 | .4356 |
| 30 | 30.55 | 634.2 | .5097 | 40 | 20.70 | 638.2 | .4709 | | | | |
| 40 | 31.20 | 639.3 | .5200 | | | | | **50** | 15.78 | 642.3 | 1.4460 |
| | | | | **50** | 21.14 | 643.4 | 1.4812 | 60 | 16.12 | 647.5 | .4562 |
| **50** | 31.85 | 644.4 | 1.5301 | 60 | 21.58 | 648.5 | .4912 | 70 | 16.45 | 652.8 | .4662 |
| 60 | 32.49 | 649.5 | .5400 | 70 | 22.01 | 653.7 | .5011 | 80 | 16.78 | 658.0 | .4760 |
| 70 | 33.14 | 654.6 | .5497 | 80 | 22.44 | 658.9 | .5108 | 90 | 17.10 | 663.2 | .4856 |
| 80 | 33.78 | 659.7 | .5593 | 90 | 22.88 | 664.0 | .5203 | | | | |
| 90 | 34.42 | 664.8 | .5687 | | | | | **100** | 17.43 | 668.5 | 1.4950 |
| | | | | **100** | 23.31 | 669.2 | 1.5296 | 110 | 17.76 | 673.7 | .5042 |
| **100** | 35.07 | 670.0 | 1.5779 | 110 | 23.74 | 674.4 | .5388 | 120 | 18.08 | 678.9 | .5133 |
| 110 | 35.71 | 675.1 | .5870 | 120 | 24.17 | 679.6 | .5478 | 130 | 18.41 | 684.2 | .5223 |
| 120 | 36.35 | 680.3 | .5960 | 130 | 24.60 | 684.8 | .5567 | 140 | 18.73 | 689.4 | .5312 |
| 130 | 36.99 | 685.4 | .6049 | 140 | 25.03 | 690.0 | .5655 | | | | |
| 140 | 37.62 | 690.6 | .6136 | | | | | **150** | 19.05 | 694.7 | 1.5399 |
| | | | | **150** | 25.46 | 695.3 | 1.5742 | 160 | 19.37 | 700.0 | .5485 |
| **150** | 38.26 | 695.8 | 1.6222 | 160 | 25.88 | 700.5 | .5827 | 170 | 19.70 | 705.3 | .5569 |
| 160 | 38.90 | 701.1 | .6307 | 170 | 26.31 | 705.8 | .5911 | 180 | 20.02 | 710.6 | .5653 |
| 170 | 39.54 | 706.3 | .6391 | 180 | 26.74 | 711.1 | .5995 | 190 | 20.34 | 715.9 | .5736 |
| 180 | 40.17 | 711.6 | .6474 | 190 | 27.16 | 716.4 | .6077 | | | | |
| 190 | 40.81 | 716.9 | .6556 | | | | | **200** | 20.66 | 721.2 | 1.5817 |
| | | | | **200** | 27.59 | 721.7 | 1.6158 | 220 | 21.30 | 732.0 | .5978 |
| **200** | 41.45 | 722.2 | 1.6637 | 220 | 28.44 | 732.4 | .6318 | 240 | 21.94 | 742.8 | .6135 |

| Temperature, Degrees F. | 25 — *7.96 Degrees* | | | Temperature, Degrees F. | 30 — *0.57 Degree* | | | Temperature, Degrees F. | 35 — *5.89 Degrees* | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | *V* | *H* | *S* | | *V* | *H* | *S* | | *V* | *H* | *S* |
| *Saturation* | *10.96* | *609.1* | *1.3515* | *Saturation* | *9.236* | *611.6* | *1.3364* | *Saturation* | *7.991* | *613.6* | *1.3236* |
| **0** | 11.19 | 613.8 | 1.3616 | 10 | 9.492 | 617.8 | 1.3497 | 10 | 8.078 | 616.1 | 1.3289 |
| 10 | 11.47 | 619.4 | .3738 | 20 | 9.731 | 623.5 | .3618 | 20 | 8.287 | 622.0 | .3413 |
| 20 | 11.75 | 625.0 | .3855 | 30 | 9.966 | 629.1 | .3733 | 30 | 8.493 | 627.7 | .3532 |
| 30 | 12.03 | 630.4 | .3967 | 40 | 10.20 | 634.6 | .3845 | 40 | 8.695 | 633.4 | .3646 |
| 40 | 12.30 | 635.8 | .4077 | | | | | | | | |
| | | | | **50** | 10.43 | 640.1 | 1.3953 | **50** | 8.895 | 638.9 | 1.3756 |
| **50** | 12.57 | 641.2 | 1.4183 | 60 | 10.65 | 645.5 | .4059 | 60 | 9.093 | 644.4 | .3863 |
| 60 | 12.84 | 646.5 | .4287 | 70 | 10.88 | 650.9 | .4161 | 70 | 9.289 | 649.9 | .3967 |
| 70 | 13.11 | 651.8 | .4388 | 80 | 11.10 | 656.2 | .4261 | 80 | 9.484 | 655.3 | .4069 |
| 80 | 13.37 | 657.1 | .4487 | 90 | 11.33 | 661.6 | .4359 | 90 | 9.677 | 660.7 | .4168 |
| 90 | 13.64 | 662.4 | .4584 | | | | | | | | |
| | | | | **100** | 11.55 | 666.9 | 1.4456 | **100** | 9.869 | 666.1 | 1.4265 |
| **100** | 13.90 | 667.7 | 1.4679 | 110 | 11.77 | 672.2 | .4550 | 110 | 10.06 | 671.5 | .4360 |
| 110 | 14.17 | 673.0 | .4772 | 120 | 11.99 | 677.5 | .4642 | 120 | 10.25 | 676.8 | .4453 |
| 120 | 14.43 | 678.2 | .4864 | 130 | 12.21 | 682.9 | .4733 | 130 | 10.44 | 682.2 | .4545 |
| 130 | 14.69 | 683.5 | .4954 | 140 | 12.43 | 688.2 | .4823 | 140 | 10.63 | 687.6 | .4635 |
| 140 | 14.95 | 688.8 | .5043 | | | | | | | | |
| | | | | **150** | 12.65 | 693.5 | 1.4911 | **150** | 10.82 | 692.9 | 1.4724 |
| **150** | 15.21 | 694.1 | 1.5131 | 160 | 12.87 | 698.8 | .4998 | 160 | 11.00 | 698.3 | .4811 |
| 160 | 15.47 | 699.4 | .5217 | 170 | 13.08 | 704.2 | .5083 | 170 | 11.19 | 703.7 | .4897 |
| 170 | 15.73 | 704.7 | .5303 | 180 | 13.30 | 709.6 | .5168 | 180 | 11.38 | 709.1 | .4982 |
| 180 | 15.99 | 710.1 | .5387 | 190 | 13.52 | 714.9 | .5251 | 190 | 11.56 | 714.5 | .5066 |
| 190 | 16.25 | 715.4 | .5470 | | | | | | | | |
| | | | | **200** | 13.73 | 720.3 | 1.5334 | **200** | 11.75 | 719.9 | 1.5148 |
| **200** | 16.50 | 720.8 | 1.5552 | 220 | 14.16 | 731.1 | .5495 | 220 | 12.12 | 730.7 | .5311 |
| 220 | 17.02 | 731.6 | .5713 | 240 | 14.59 | 742.0 | .5653 | 240 | 12.49 | 741.7 | .5469 |
| 240 | 17.53 | 742.5 | .5870 | 260 | 15.02 | 753.0 | .5808 | 260 | 12.86 | 752.7 | .5624 |
| 260 | 18.04 | 753.4 | .6025 | 280 | 15.45 | 764.1 | .5960 | 280 | 13.23 | 763.7 | .5776 |

TABLE 55b—*Continued*

| Temperature, Degrees F. | 40 *11.66 Degrees* | | | Temperature, Degrees F. | 50 *21.67 Degrees* | | | Temperature, Degrees F. | 60 *30.21 Degrees* | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | *V* | *H* | *S* | | *V* | *H* | *S* | | *V* | *H* | *S* |
| *Saturation* | *7.047* | *615.4* | *1.3125* | *Saturation* | *5.710* | *618.2* | *1.2939* | *Saturation* | *4.805* | *620.5* | *1.2787* |
| 20 | 7.203 | 620.4 | 1.3231 | 30 | 5.838 | 623.4 | 1.3046 | 40 | 4.933 | 626.8 | 1.2913 |
| 30 | 7.387 | 626.3 | .3353 | 40 | 5.988 | 629.5 | 3169 | | | | |
| 40 | 7.568 | 632.1 | .3470 | | | | | **50** | 5.060 | 632.9 | 1.3035 |
| | | | | **50** | 6.135 | 635.4 | 1.3286 | 60 | 5.184 | 639.0 | .3152 |
| **50** | 7.746 | 637.8 | 1.3583 | 60 | 6.280 | 641.2 | .3399 | 70 | 5.307 | 644.9 | .3265 |
| 60 | 7.922 | 643.4 | .3692 | 70 | 6.423 | 646.9 | .3508 | 80 | 5.428 | 650.7 | .3373 |
| 70 | 8.096 | 648.9 | .3797 | 80 | 6.564 | 652.6 | .3613 | 90 | 5.547 | 656.4 | .3479 |
| 80 | 8.268 | 654.4 | .3900 | 90 | 6.704 | 658.2 | .3716 | | | | |
| 90 | 8.439 | 659.9 | .4000 | | | | | **100** | 5.665 | 662.1 | 1.3581 |
| | | | | **100** | 6.843 | 663.7 | 1.3816 | 110 | 5.781 | 667.7 | .3681 |
| **100** | 8.609 | 665.3 | 1.4098 | 110 | 6.980 | 669.2 | .3914 | 120 | 5.897 | 673.3 | .3778 |
| 110 | 8.777 | 670.7 | .4194 | 120 | 7.117 | 674.7 | .4009 | 130 | 6.012 | 678.9 | .3873 |
| 120 | 8.945 | 676.1 | .4288 | 130 | 7.252 | 680.2 | .4103 | 140 | 6.126 | 684.4 | .3966 |
| 130 | 9.112 | 681.5 | .4381 | 140 | 7.387 | 685.7 | 4195 | | | | |
| 140 | 9.278 | 686.9 | .4471 | | | | | **150** | 6.239 | 689.9 | 1.4058 |
| | | | | **150** | 7.521 | 691.1 | 1.4286 | 160 | 6.352 | 695.5 | .4148 |
| **150** | 9.444 | 692.3 | 1.4561 | 160 | 7.655 | 696.6 | .4374 | 170 | 6.464 | 701.0 | .4236 |
| 160 | 9.609 | 697.7 | .4648 | 170 | 7.788 | 702.1 | .4462 | 180 | 6.576 | 706.5 | .4323 |
| 170 | 9.774 | 703.1 | .4735 | 180 | 7.921 | 707.5 | .4548 | 190 | 6.687 | 712.0 | .4409 |
| 180 | 9.938 | 708.5 | .4820 | 190 | 8.053 | 713.0 | .4633 | | | | |
| 190 | 10.10 | 714.0 | .4904 | | | | | **200** | 6.798 | 717.5 | 1.4493 |
| | | | | **200** | 8.185 | 718.5 | 1.4716 | 210 | 6.909 | 723.1 | .4576 |
| **200** | 10.27 | 719.4 | 1.4987 | 210 | 8.317 | 724.0 | .4799 | 220 | 7.019 | 728.6 | .4658 |
| 220 | 10.59 | 730.3 | .5150 | 220 | 8.448 | 729.4 | .4880 | 230 | 7.129 | 734.1 | .4739 |
| 240 | 10.92 | 741.3 | .5309 | 240 | 8.710 | 740.5 | .5040 | 240 | 7.238 | 739.7 | .4819 |
| 260 | 11.24 | 752.3 | .5465 | 260 | 8.970 | 751.6 | .5197 | | | | |
| 280 | 11.56 | 763.4 | .5617 | | | | | **260** | 7.457 | 750.9 | 1.4976 |
| | | | | **280** | 9.230 | 762.7 | 1.5350 | 280 | 7.675 | 762.1 | .5130 |
| **300** | 11.88 | 774.6 | .5766 | 300 | 9.489 | 774.0 | .5500 | 300 | 7.892 | 773.3 | .5281 |

| Temperature, Degrees F. | 75 *41.13 Degrees* | | | Temperature, Degrees F. | 100 *56.05 Degrees* | | | Temperature, Degrees F. | 125 *68.31 Degrees* | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | *V* | *H* | *S* | | *V* | *H* | *S* | | *V* | *H* | *S* |
| *Saturation* | *3.887* | *623.2* | *1.2599* | *Saturation* | *2.952* | *626.5* | *1.2356* | *Saturation* | *2.380* | *628.8* | *1.2166* |
| **50** | 3.982 | 629.1 | 1.2715 | 70 | 3.068 | 636.0 | 1.2539 | 80 | 2.461 | 637.2 | 1.2322 |
| 60 | 4.087 | 635.5 | .2839 | 80 | 3.149 | 642.6 | .2661 | 90 | 2.528 | 644.0 | .2448 |
| 70 | 4.189 | 641.7 | .2957 | 90 | 3.227 | 649.0 | .2778 | | | | |
| 80 | 4.289 | 647.7 | .3071 | | | | | **100** | 2.593 | 650.7 | 1.2568 |
| 90 | 4.388 | 653.7 | .3180 | **100** | 3.304 | 655.2 | 1.2891 | 110 | 2.657 | 657.1 | .2682 |
| | | | | 110 | 3.380 | 661.3 | .2999 | 120 | 2.719 | 663.5 | .2792 |
| **100** | 4.485 | 659.6 | 1.3286 | 120 | 3.454 | 667.3 | .3104 | 130 | 2.780 | 669.7 | .2899 |
| 110 | 4.581 | 665.4 | .3389 | 130 | 3.527 | 673.3 | .3206 | 140 | 2.840 | 675.8 | .3002 |
| 120 | 4.676 | 671.1 | .3489 | 140 | 3.600 | 679.2 | .3305 | | | | |
| 130 | 4.770 | 676.8 | .3586 | | | | | **150** | 2.900 | 681.8 | 1.3102 |
| 140 | 4.863 | 682.5 | .3682 | **150** | 3.672 | 685.0 | 1.3401 | 160 | 2.958 | 687.8 | .3199 |
| | | | | 160 | 3.743 | 690.8 | .3495 | 170 | 3.016 | 693.7 | .3294 |
| **150** | 4.956 | 688.1 | 1.3775 | 170 | 3.813 | 696.6 | .3588 | 180 | 3.074 | 699.6 | .3387 |
| 160 | 5.048 | 693.7 | .3866 | 180 | 3.883 | 702.3 | .3678 | 190 | 3.131 | 705.5 | .3478 |
| 170 | 5.139 | 699.3 | .3956 | 190 | 3.952 | 708.0 | .3767 | | | | |
| 180 | 5.230 | 704.9 | .4044 | | | | | **200** | 3.187 | 711.3 | 1.3567 |
| 190 | 5.320 | 710.5 | .4131 | **200** | 4.021 | 713.7 | 1.3854 | 210 | 3.243 | 717.2 | .3654 |
| | | | | 210 | 4.090 | 719.4 | .3940 | 220 | 3.299 | 723.0 | .3740 |
| **200** | 5.410 | 716.1 | 1.4217 | 220 | 4.158 | 725.1 | .4024 | 230 | 3.354 | 728.8 | .3825 |
| 210 | 5.500 | 721.7 | .4301 | 230 | 4.226 | 730.8 | .4108 | 240 | 3.409 | 734.5 | .3908 |
| 220 | 5.589 | 727.3 | .4384 | 240 | 4.294 | 736.5 | .4190 | | | | |
| 230 | 5.678 | 732.9 | .4466 | | | | | **250** | 3.464 | 740.3 | 1.3990 |
| 240 | 5.767 | 738.5 | .4546 | **250** | 4.361 | 742.2 | 1.4271 | 260 | 3.519 | 746.1 | .4071 |
| | | | | 260 | 4.428 | 747.9 | .4350 | 270 | 3.573 | 751.9 | .4151 |
| **250** | 5.855 | 744.1 | 1.4625 | 270 | 4.495 | 753.6 | .4429 | 280 | 3.627 | 757.7 | .4230 |
| 260 | 5.943 | 749.8 | .4705 | 280 | 4.562 | 759.4 | .4507 | 290 | 3.681 | 763.5 | .4308 |
| 280 | 6.119 | 761.1 | .4860 | 290 | 4.629 | 765.1 | .4584 | | | | |
| 300 | 6.294 | 772.4 | .5011 | | | | | **300** | 3.735 | 769.3 | 1.4385 |
| | | | | **300** | 4.695 | 770.8 | 1.4660 | 320 | 3.842 | 780.9 | .4536 |

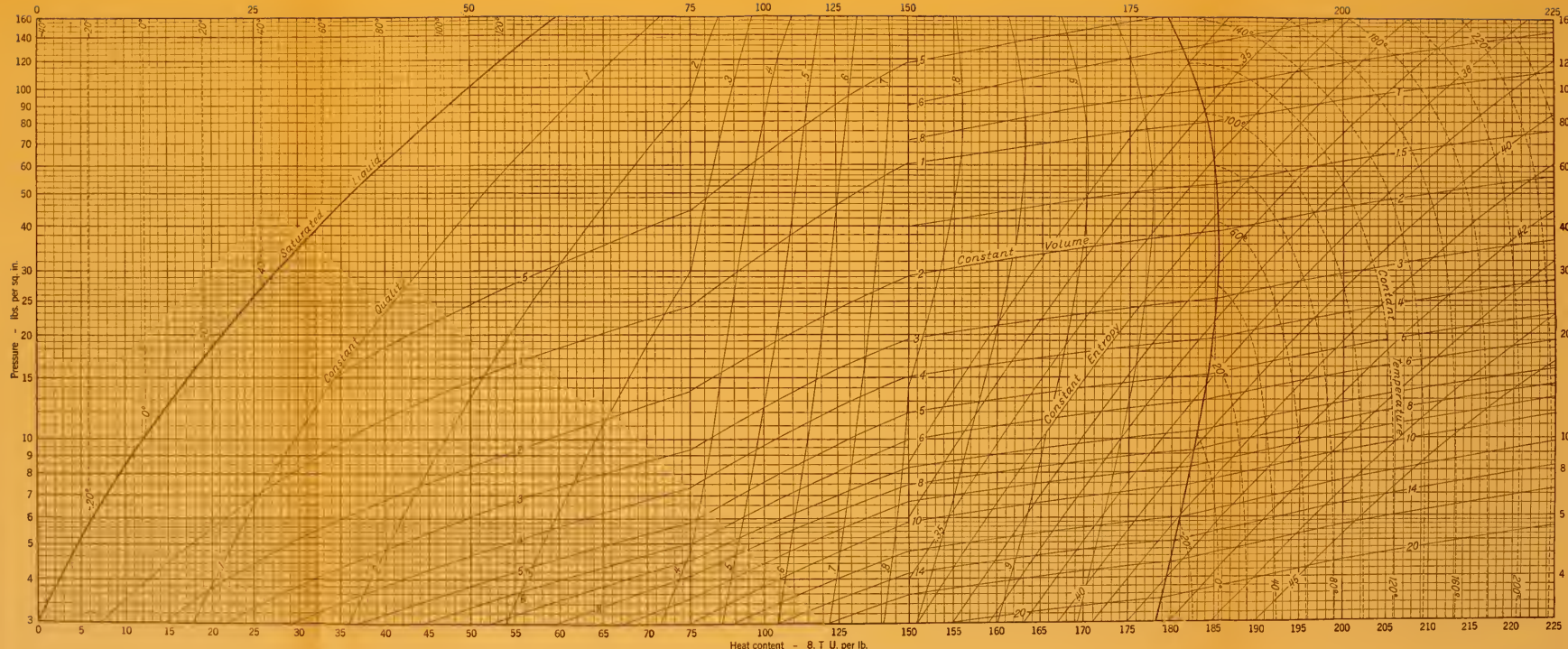



FIG. 166.—P–I Diagram for Sulphur Dioxide.

To face page 235.

TABLE 55*b*—*Continued*

| Temperature, Degrees F. | 150 *78.81 Degrees* | | | Temperature, Degrees F. | 200 *96.34 Degrees* | | | Temperature, Degrees F. | 250 *110.80 Degrees* | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | *V* | *H* | *S* | | *V* | *H* | *S* | | *V* | *H* | *S* |
| *Saturation* | *1.994* | *630.5* | *1.2009* | *Saturation* | *1.502* | *632.7* | *1.1756* | *Saturation* | *1.202* | *633.8* | *1.1555* |
| 90 | 2.061 | 638.8 | 1.2161 | 110 | 1.567 | 643.4 | 1.1947 | 120 | 1.240 | 641.5 | 1.1690 |
| | | | | 120 | 1.612 | 650.9 | .2077 | 130 | 1.278 | 649.6 | .1827 |
| **100** | 2.118 | 645.9 | 1.2289 | 130 | 1.656 | 658.1 | .2200 | 140 | 1.316 | 657.2 | .1956 |
| 110 | 2.174 | 652.8 | .2410 | 140 | 1.698 | 665.0 | .2317 | | | | |
| 120 | 2.228 | 659.4 | .2526 | | | | | **150** | 1.352 | 664.6 | 1.2078 |
| 130 | 2.281 | 665.9 | .2638 | **150** | 1.740 | 671.8 | 1.2429 | 160 | 1.386 | 671.8 | .2195 |
| 140 | 2.334 | 672.3 | .2745 | 160 | 1.780 | 678.4 | .2537 | 170 | 1.420 | 678.7 | .2306 |
| | | | | 170 | 1.820 | 684.9 | .2641 | 180 | 1.453 | 685.5 | .2414 |
| **150** | 2.385 | 678.6 | 1.2849 | 180 | 1.859 | 691.3 | .2742 | 190 | 1.486 | 692.2 | .2517 |
| 160 | 2.435 | 684.8 | .2949 | 190 | 1.897 | 697.7 | .2840 | | | | |
| 170 | 2.485 | 690.9 | .3047 | | | | | **200** | 1.518 | 698.8 | 1.2617 |
| 180 | 2.534 | 696.9 | .3142 | **200** | 1.935 | 703.9 | 1.2935 | 210 | 1.549 | 705.3 | .2715 |
| 190 | 2.583 | 702.9 | .3236 | 210 | 1.972 | 710.1 | .3029 | 220 | 1.580 | 711.7 | .2810 |
| | | | | 220 | 2.009 | 716.3 | .3120 | 230 | 1.610 | 718.0 | .2902 |
| **200** | 2.631 | 708.9 | 1.3327 | 230 | 2.046 | 722.4 | .3209 | 240 | 1.640 | 724.3 | .2993 |
| 210 | 2.679 | 714.8 | .3416 | 240 | 2.082 | 728.4 | .3296 | | | | |
| 220 | 2.726 | 720.7 | .3504 | | | | | **250** | 1.670 | 730.5 | 1.3081 |
| 230 | 2.773 | 726.6 | .3590 | **250** | 2.118 | 734.5 | 1.3382 | 260 | 1.699 | 736.7 | .3168 |
| 240 | 2.820 | 732.5 | .3675 | 260 | 2.154 | 740.5 | .3467 | 270 | 1.729 | 742.9 | .3253 |
| | | | | 270 | 2.189 | 746.5 | .3550 | 280 | 1.758 | 749.1 | .3337 |
| **250** | 2.866 | 738.4 | 1.3758 | 280 | 2.225 | 752.5 | .3631 | 290 | 1.786 | 755.2 | .3420 |
| 260 | 2.912 | 744.3 | .3840 | 290 | 2.260 | 758.5 | .3712 | | | | |
| 270 | 2.958 | 750.1 | .3921 | | | | | **300** | 1.815 | 761.3 | 1.3501 |
| 280 | 3.004 | 756.0 | .4001 | **300** | 2.295 | 764.5 | 1.3791 | 320 | 1.872 | 773.5 | .3659 |
| 290 | 3.049 | 761.8 | .4079 | 320 | 2.364 | 776.5 | .3947 | 340 | 1.928 | 785.7 | .3814 |
| | | | | 340 | 2.432 | 788.5 | .4099 | 360 | 1.983 | 797.9 | .3964 |
| **300** | 3.095 | 767.7 | 1.4157 | 360 | 2.500 | 800.5 | .4247 | 380 | 2.038 | 810.1 | .4111 |
| 320 | 3.185 | 779.4 | .4310 | 380 | 2.568 | 812.5 | .4392 | | | | |
| 340 | 3.274 | 791.2 | .4459 | | | | | **400** | 2.093 | 822.3 | 1.4255 |

ing properties of sulphur dioxide, an oil has to be used and this needs to be absolutely anhydrous or corrosion will develop. Table 56 gives the thermodynamic properties of sulphur dioxide.

**Ethyl Chloride.**—Ethyl chloride ($C_2H_5Cl$) is a colorless liquid, slightly lighter in weight than water, which may be stored in moderate pressure containers. It is an anæsthetic but must be inhaled in considerable quantities in order to have an appreciable effect. It is very inflammable and can be exploded by a spark at atmospheric pressures in mixtures of gas in air of not less than 4.9 and not greater than 13.5 per cent by volume. The corresponding mixtures for ammonia and gasoline are 16.5 to 26.8 per cent and 1.5 to 6.4 per cent respectively. It does not corrode iron,[4] nor copper or its alloys, nor aluminum. The condensing pressure is very small—about 27 lb. abs. at 86 deg. F. (Table 57), and the evaporating pressure is usually less than one atmosphere—and the piston displacement per ton of refrigeration is very high, some 30 times that for carbon dioxide and six times that of ammonia. Because of the low pressures and the large volumes to be compressed per unit of refrigeration, the ethyl chloride compressor has been designed as a rotary

[4] Special Report No. 14, Flood Investigation Board, by Jenkin and Shorthose.

## TABLE 56

### Saturated Sulphur Dioxide: Temperature Table

| Temperature, Degrees F. $t$ | Pressure (Absolute), Pounds per Inch$^2$ $p$ | Volume, Foot$^3$-Pounds | | Heat Content, B.t.u. per Pound | | Latent Heat, B.t.u. per Pound | | Entropy | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | Liquid $v'$ | Vapor $v''$ | Liquid $q$ | Vapor $i''$ | Total $r$ | Internal | Liquid $s'$ | $\frac{r}{T}$ | Vapor $s''$ |
| −40 | 3.136 | .010440 | 22.42 | 0.00 | 178.61 | 178.61 | 165.57 | .0000000 | .42562 | .42562 |
| −35 | 3.693 | .010486 | 19.23 | 1.45 | 179.27 | 177.82 | 164.66 | .00334 | .41875 | .42209 |
| −30 | 4.331 | .010532 | 16.56 | 2.93 | 179.90 | 176.97 | 163.70 | .00674 | .41190 | .41864 |
| −25 | 5.058 | .010580 | 14.31 | 4.44 | 180.50 | 176.06 | 162.67 | .01016 | .40509 | .41525 |
| −20 | 5.883 | .010627 | 12.42 | 5.98 | 181.07 | 175.09 | 161.58 | .01366 | .39826 | .41192 |
| −15 | 6.814 | .010674 | 10.81 | 7.56 | 181.62 | 174.06 | 160.44 | .01719 | .39146 | .40865 |
| −10 | 7.863 | .010721 | 9.44 | 9.16 | 182.13 | 172.97 | 159.24 | .02075 | .38469 | .40544 |
| − 5 | 9.038 | .010770 | 8.28 | 10.79 | 182.62 | 171.83 | 157.29 | .02443 | .37795 | .40228 |
| 0 | 10.35 | .010820 | 7.280 | 12.44 | 183.07 | 170.63 | 156.70 | .02795 | .37122 | .39917 |
| 1 | 10.63 | .010830 | 7.099 | 12.79 | 183.17 | 170.38 | 156.44 | .02869 | .36987 | .39856 |
| 2 | 10.91 | .010840 | 6.923 | 13.12 | 183.25 | 170.13 | 156.17 | .02941 | .36853 | .39794 |
| 3 | 11.20 | .010850 | 6.751 | 13.45 | 183.33 | 169.88 | 155.90 | .03013 | .36719 | .39732 |
| 4 | 11.50 | .010860 | 6.584 | 13.78 | 183.41 | 169.63 | 155.63 | .03084 | .36586 | .39670 |
| 5 | 11.81 | .010870 | 6.421 | 14.11 | 183.49 | 169.38 | 155.36 | .03155 | .36454 | .39609 |
| 6 | 12.12 | .010880 | 6.266 | 14.45 | 183.57 | 169.12 | 155.09 | .03228 | .36319 | .39547 |
| 7 | 12.43 | .010890 | 6.114 | 14.79 | 183.65 | 168.86 | 154.81 | .03300 | .36186 | .39486 |
| 8 | 12.75 | .010900 | 5.967 | 15.13 | 183.73 | 168.60 | 154.54 | .03373 | .36053 | .39426 |
| 9 | 13.08 | .010910 | 5.822 | 15.46 | 183.80 | 168.34 | 154.26 | .03445 | .35921 | .39366 |
| 10 | 13.42 | .010920 | 5.682 | 15.80 | 183.87 | 168.07 | 153.98 | .03519 | .35787 | .39306 |
| 11 | 13.77 | .010930 | 5.548 | 16.14 | 183.94 | 167.80 | 153.70 | .03592 | .35654 | .39246 |
| 12 | 14.12 | .010940 | 5.417 | 16.48 | 184.01 | 167.53 | 153.42 | .03664 | .35521 | .39185 |
| 13 | 14.48 | .010950 | 5.289 | 16.81 | 184.07 | 167.26 | 153.13 | .03737 | .35388 | .39125 |
| 14 | 14.84 | .010960 | 5.164 | 17.15 | 184.14 | 166.97 | 152.84 | .03808 | .35257 | .39065 |
| 15 | 15.21 | .010971 | 5.042 | 17.49 | 184.21 | 166.72 | 152.55 | .03880 | .35125 | .39005 |
| 16 | 15.59 | .010981 | 4.926 | 17.84 | 184.28 | 166.44 | 152.26 | .03953 | .34993 | .38946 |
| 17 | 15.98 | .010992 | 4.812 | 18.18 | 184.34 | 166.16 | 151.97 | .04026 | .34861 | .38887 |
| 18 | 16.37 | .011003 | 4.701 | 18.52 | 184.40 | 165.88 | 151.68 | .04098 | .34729 | .38827 |
| 19 | 16.77 | .011014 | 4.593 | 18.86 | 184.46 | 165.60 | 151.38 | .04169 | .34598 | .38767 |
| 20 | 17.18 | .011025 | 4.487 | 19.20 | 184.52 | 165.32 | 151.08 | .04241 | .34466 | .38707 |
| 21 | 17.60 | .011036 | 4.386 | 19.55 | 184.58 | 165.03 | 150.78 | .04313 | .34335 | .38648 |
| 22 | 18.03 | .011047 | 4.287 | 19.90 | 184.64 | 164.74 | 150.48 | .04385 | .34204 | .38589 |
| 23 | 18.46 | .011058 | 4.190 | 20.24 | 184.69 | 164.45 | 150.18 | .04457 | .34073 | .38530 |
| 24 | 18.89 | .011070 | 4.096 | 20.58 | 184.74 | 164.16 | 149.88 | .04528 | .33943 | .38471 |
| 25 | 19.34 | .011082 | 3.994 | 20.92 | 184.79 | 163.87 | 149.57 | .04600 | .33812 | .38412 |
| 26 | 19.80 | .011093 | 3.915 | 21.26 | 184.84 | 163.58 | 149.27 | .04671 | .33683 | .38354 |
| 27 | 20.26 | .011104 | 3.829 | 21.61 | 184.89 | 163.28 | 148.97 | .04743 | .33553 | .38296 |
| 28 | 20.73 | .011116 | 3.744 | 21.96 | 184.94 | 162.98 | 148.66 | .04814 | .33422 | .38237 |
| 29 | 21.21 | .011128 | 3.662 | 22.30 | 184.98 | 162.68 | 148.34 | .04886 | .33292 | .38178 |
| 30 | 21.70 | .011140 | 3.581 | 22.64 | 185.02 | 162.38 | 148.02 | .04956 | .33163 | .38119 |
| 31 | 22.20 | .011152 | 3.503 | 22.98 | 185.06 | 162.08 | 147.72 | .05027 | .33034 | .38061 |
| 32 | 22.71 | .011164 | 3.437 | 23.33 | 185.10 | 161.77 | 147.41 | .05099 | .32904 | .38003 |
| 33 | 23.23 | .011176 | 3.355 | 23.68 | 185.14 | 161.46 | 147.09 | .05171 | .32774 | .37945 |
| 34 | 23.75 | .011188 | 3.283 | 24.03 | 185.18 | 161.15 | 146.77 | .05242 | .32645 | .37887 |
| 35 | 24.28 | .011200 | 3.212 | 24.38 | 185.22 | 160.84 | 146.45 | .05312 | .32517 | .37829 |
| 36 | 24.82 | .011212 | 3.144 | 24.72 | 185.25 | 160.53 | 146.13 | .05384 | .32388 | .37772 |
| 37 | 25.39 | .011224 | 3.078 | 25.07 | 185.28 | 160.21 | 145.81 | .05456 | .32259 | .37715 |
| 38 | 25.95 | .011236 | 3.013 | 25.42 | 185.31 | 159.89 | 145.48 | .05527 | .32130 | .37657 |
| 39 | 26.52 | .011248 | 2.949 | 25.77 | 185.34 | 159.57 | 145.15 | .05598 | .32001 | .37599 |
| 40 | 27.10 | .011260 | 2.887 | 26.12 | 185.37 | 159.25 | 144.82 | .05668 | .31873 | .37541 |
| 41 | 27.69 | .011272 | 2.827 | 26.47 | 185.40 | 158.93 | 144.55 | .05738 | .31745 | .47483 |
| 42 | 28.29 | .011284 | 2.769 | 26.81 | 185.42 | 158.61 | 144.17 | .05809 | .31616 | .37425 |
| 43 | 28.90 | .011296 | 2.712 | 27.16 | 185.44 | 158.28 | 144.84 | .05879 | .31489 | .37368 |
| 44 | 29.52 | .011308 | 2.656 | 27.51 | 185.46 | 157.95 | 144.50 | .05949 | .31362 | .37311 |
| 45 | 30.15 | .011320 | 2.601 | 27.86 | 185.48 | 157.62 | 143.16 | .06020 | .31234 | .37254 |
| 46 | 30.79 | .011332 | 2.548 | 28.21 | 185.50 | 157.29 | 143.83 | .06090 | .31107 | .37197 |
| 47 | 31.44 | .011344 | 2.497 | 28.56 | 185.52 | 156.96 | 142.49 | .06161 | .30979 | .37140 |
| 48 | 32.10 | .011356 | 2.446 | 28.92 | 185.54 | 156.62 | 142.15 | .06231 | .30852 | .37083 |
| 49 | 32.77 | .011368 | 2.397 | 29.27 | 185.55 | 156.28 | 142.81 | .06301 | .30725 | .37026 |

TABLE 56—*Continued*

| Temperature, Degrees F. $t$ | Pressure (Absolute), Pounds per Inch² $p$ | Volume, Foot³-Pounds | | Heat Content, B.t.u. per Pound | | Latent Heat, B.t.u. per Pound | | Entropy | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | Liquid $v'$ | Vapor $v''$ | Liquid $q$ | Vapor $i''$ | Total $r$ | Internal | Liquid $s'$ | $\frac{r}{T}$ | Vapor $s''$ |
| 50 | 33.45 | .011380 | 2.348 | 29.61 | 185.56 | 155.95 | 141.47 | .06370 | .30599 | .36969 |
| 51 | 34.15 | .011392 | 2.302 | 29.96 | 185.57 | 159.61 | 141.13 | .06439 | .30474 | .36913 |
| 52 | 34.86 | .011404 | 2.256 | 30.31 | 185.58 | 155.27 | 140.79 | .06509 | .30348 | .36857 |
| 53 | 35.58 | .011416 | 2.211 | 30.66 | 185.59 | 154.93 | 140.44 | .06578 | .30222 | .36800 |
| 54 | 36.31 | .011428 | 2.167 | 31.00 | 185.59 | 154.59 | 140.09 | .06646 | .30097 | .36743 |
| 55 | 37.05 | .011440 | 2.124 | 31.36 | 185.60 | 154.24 | 139.74 | .06715 | .29971 | .36686 |
| 56 | 37.80 | .011452 | 2.083 | 31.72 | 185.61 | 153.89 | 139.39 | .06785 | .29844 | .36629 |
| 57 | 38.56 | .011464 | 2.043 | 32.08 | 185.62 | 153.54 | 139.04 | .06854 | .29719 | .36573 |
| 58 | 39.33 | .011476 | 2.003 | 32.42 | 185.61 | 153.19 | 138.69 | .06923 | .29594 | .36517 |
| 59 | 40.12 | .011488 | 1.964 | 32.76 | 185.60 | 152.84 | 138.34 | .06992 | .29469 | .36461 |
| 60 | 40.93 | .011500 | 1.926 | 33.10 | 185.59 | 152.49 | 137.98 | .07060 | .29345 | .36405 |
| 61 | 41.75 | .011512 | 1.889 | 33.44 | 185.58 | 152.14 | 137.53 | .07128 | .29221 | .36349 |
| 62 | 42.58 | .011524 | 1.853 | 33.79 | 185.57 | 151.78 | 137.17 | .07196 | .29097 | .36293 |
| 63 | 43.42 | .011536 | 1.816 | 34.14 | 185.56 | 151.42 | 136.81 | .07265 | .28972 | .36237 |
| 64 | 44.27 | .011548 | 1.783 | 34.49 | 185.55 | 151.06 | 136.35 | .07333 | .28848 | .36181 |
| 65 | 45.13 | .011560 | 1.749 | 34.84 | 185.54 | 150.70 | 136.19 | .07401 | .28724 | .36125 |
| 66 | 46.00 | .011572 | 1.716 | 35.19 | 185.53 | 150.34 | 135.83 | .07469 | .28601 | .36070 |
| 67 | 46.88 | .011585 | 1.683 | 35.54 | 185.52 | 149.98 | 135.47 | .07535 | .28479 | .36014 |
| 68 | 47.78 | .011598 | 1.652 | 35.88 | 185.50 | 149.62 | 135.11 | .07602 | .28356 | .35958 |
| 69 | 48.69 | .011611 | 1.621 | 36.23 | 185.48 | 149.25 | 134.74 | .07769 | .28233 | .35902 |
| 70 | 49.62 | .011626 | 1.590 | 36.58 | 185.46 | 148.88 | 134.37 | .07736 | .28110 | .35846 |
| 71 | 50.57 | .011639 | 1.557 | 36.93 | 185.44 | 148.51 | 134.01 | .07804 | .27987 | .35791 |
| 72 | 51.54 | .011652 | 1.532 | 37.28 | 185.42 | 148.14 | 133.64 | .07871 | .27865 | .35736 |
| 73 | 52.51 | .011666 | 1.503 | 37.63 | 185.40 | 147.77 | 133.27 | .07937 | .27743 | .35680 |
| 74 | 53.48 | .011680 | 1.476 | 37.97 | 185.37 | 147.40 | 132.90 | .08003 | .27621 | .35624 |
| 75 | 54.47 | .011693 | 1.448 | 38.32 | 185.34 | 147.02 | 132.53 | .08070 | .27498 | .35568 |
| 76 | 55.48 | .011706 | 1.422 | 38.67 | 185.31 | 146.64 | 132.16 | .08135 | .27377 | .35512 |
| 77 | 56.51 | .011719 | 1.396 | 39.01 | 185.27 | 146.26 | 131.79 | .08201 | .27255 | .35456 |
| 78 | 57.56 | .011732 | 1.371 | 39.36 | 185.24 | 145.88 | 131.41 | .08268 | .27133 | .35401 |
| 79 | 58.62 | .011746 | 1.343 | 39.71 | 185.21 | 145.50 | 131.03 | .08336 | .27012 | .35346 |
| 80 | 59.68 | .011760 | 1.321 | 40.05 | 185.17 | 145.12 | 130.65 | .08399 | .26897 | .35291 |
| 81 | 60.77 | .011773 | 1.297 | 40.39 | 185.13 | 144.74 | 130.27 | .08462 | .26772 | .35234 |
| 82 | 61.88 | .011786 | 1.274 | 40.73 | 185.09 | 144.36 | 129.89 | .08525 | .26652 | .35177 |
| 83 | 62.01 | .011800 | 1.253 | 41.08 | 185.05 | 143.97 | 129.51 | .08589 | .26532 | .35121 |
| 84 | 64.14 | .011814 | 1.229 | 41.43 | 185.01 | 143.58 | 129.13 | .08653 | .26412 | .35065 |
| 85 | 65.28 | .011828 | 1.207 | 41.78 | 184.97 | 143.19 | 128.75 | .08718 | .26291 | .35009 |
| 86 | 66.45 | .011841 | 1.185 | 42.12 | 184.92 | 142.80 | 128.37 | .08783 | .26171 | .34954 |
| 87 | 67.64 | .011854 | 1.164 | 42.46 | 184.87 | 142.41 | 127.99 | .08847 | .26052 | .34899 |
| 88 | 68.84 | .011868 | 1.144 | 42.80 | 184.82 | 142.02 | 127.60 | .08910 | .25933 | .34843 |
| 89 | 70.04 | .011882 | 1.124 | 43.15 | 184.77 | 141.62 | 127.21 | .08974 | .25813 | .34787 |
| 90 | 71.25 | .011896 | 1.104 | 43.50 | 184.72 | 141.22 | 126.82 | .09038 | .25693 | .34731 |
| 91 | 72.46 | .011909 | 1.084 | 43.85 | 184.67 | 140.82 | 126.43 | .09102 | .25574 | .34676 |
| 92 | 73.70 | .011923 | 1.065 | 44.19 | 184.61 | 140.42 | 125.04 | .09165 | .25455 | .34620 |
| 93 | 74.98 | .011937 | 1.047 | 44.53 | 184.55 | 140.02 | 125.65 | .09227 | .25337 | .34564 |
| 94 | 76.30 | .011951 | 1.028 | 44.86 | 184.49 | 139.62 | 125.26 | .09389 | .25219 | .34508 |
| 95 | 77.60 | .011965 | 1.011 | 45.20 | 184.43 | 139.23 | 124.87 | .09349 | .25103 | .34452 |
| 96 | 79.03 | .011979 | .9931 | 45.54 | 184.37 | 138.83 | 124.48 | .09411 | .24986 | .34397 |
| 97 | 80.40 | .011993 | .9759 | 45.88 | 184.31 | 138.43 | 124.09 | .09472 | .24869 | .34341 |
| 98 | 81.77 | .012008 | .9591 | 46.22 | 184.25 | 138.03 | 123.69 | .09532 | .24753 | .34285 |
| 99 | 82.14 | .012002 | .9425 | 46.56 | 184.18 | 137.62 | 123.29 | .09557 | .24635 | .34229 |
| 100 | 84.52 | .012037 | .9262 | 46.90 | 184.10 | 137.20 | 122.89 | .09657 | .24516 | .34173 |
| 105 | 91.85 | .012110 | .8498 | 48.88 | 183.72 | 135.14 | 120.88 | .09958 | .23934 | .33892 |
| 110 | 99.76 | .012190 | .7804 | 50.26 | 183.31 | 133.05 | 118.86 | .10254 | .23357 | .33611 |
| 115 | 108.02 | .012275 | .7174 | 51.93 | 182.85 | 130.92 | 116.82 | .10546 | .22783 | .33329 |
| 120 | 120.93 | .012360 | .6598 | 53.58 | 182.36 | 128.78 | 114.70 | .10829 | .22217 | .33046 |
| 125 | 126.43 | .012445 | .6079 | 55.31 | 181.82 | 126.51 | 112.67 | .11120 | .21639 | .32759 |
| 130 | 136.48 | .012530 | .5595 | 56.85 | 181.24 | 124.39 | 110.57 | .11376 | .21096 | .32472 |
| 135 | 147.21 | .012620 | .5158 | 58.47 | 180.62 | 122.15 | 108.45 | .11640 | .20542 | .32182 |
| 140 | 158.61 | .012720 | .4758 | 60.04 | 179.94 | 119.90 | 106.30 | .11893 | .19990 | .31888 |

## TABLE 56*a*

### Saturated Sulphur Dioxide: Pressure Table

| Pressure, Pound-inch² Absolute $p$ | Temperature, Degrees F. $t$ | Volume of Vapor, Foot³-pound $v''$ | Heat, B.t.u. per Pound | | Entropy | |
|---|---|---|---|---|---|---|
| | | | Liquid $q$ | Vapor $i''$ | Liquid $s'$ | Vapor $s''$ |
| 5 | −25.40 | 14.47 | 4.32 | 180.45 | .00988 | .41552 |
| 10 | − 1.34 | 7.520 | 12.00 | 182.95 | .02698 | .40000 |
| 15 | 14.43 | 5.110 | 17.30 | 184.17 | .03839 | .39091 |
| 20 | 26.44 | 3.878 | 21.41 | 184.86 | .04702 | .38329 |
| 25 | 36.33 | 3.123 | 24.83 | 185.26 | .05407 | .37754 |
| 30 | 44.76 | 2.614 | 27.78 | 185.48 | .06003 | .37269 |
| 35 | 52.20 | 2.247 | 30.38 | 185.58 | .06522 | .36848 |
| 40 | 58.83 | 1.970 | 32.73 | 185.60 | .06982 | .36470 |
| 45 | 64.85 | 1.754 | 34.79 | 185.54 | .07390 | .36133 |
| 50 | 70.40 | 1.577 | 36.72 | 185.45 | .07763 | .35826 |
| 55 | 75.53 | 1.434 | 38.51 | 185.32 | .08105 | .35539 |
| 60 | 80.29 | 1.314 | 40.15 | 185.16 | .08418 | .35373 |
| 65 | 84.76 | 1.211 | 41.69 | 184.98 | .08705 | .35023 |
| 70 | 88.97 | 1.125 | 43.14 | 184.77 | .08975 | .34789 |
| 75 | 93.00 | 1.047 | 44.50 | 184.55 | .09228 | .34568 |
| 80 | 96.88 | .9809 | 45.78 | 184.33 | .09464 | .34557 |

## TABLE 56*b*

### Properties of Superheated Sulphur Dioxide

Absolute Pressure in Pounds per Square inch

| Temperature, Degrees F. | 4 *−32.60 Degrees* | | | Temperature, Degrees F. | 6 *−19.37 Degrees* | | | Temperature, Degrees F. | 8 *−8.99 Degrees* | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | $v$ | $i$ | $s$ | | $v$ | $i$ | $s$ | | $v$ | $i$ | $s$ |
| *Saturation* | *17.30* | *179.57* | *.42043* | *Saturation* | *12.22* | *181.14* | *.41151* | *Saturation* | *9.220* | *182.23* | *.40482* |
| −20 | 18.40 | 181.5 | .42487 | **0** | 12.75 | 184.3 | .41850 | **0** | 9.516 | 183.7 | .40871 |
| −10 | 18.83 | 183.0 | .42836 | 10 | 13.04 | 185.9 | .42198 | 10 | 9.751 | 185.4 | .41230 |
| | | | | 20 | 13.34 | 187.5 | .42538 | 20 | 9.983 | 187.1 | .41579 |
| **0** | 19.27 | 184.6 | .43179 | 30 | 13.63 | 189.1 | .42869 | 30 | 10.21 | 188.8 | .41922 |
| 10 | 19.70 | 186.1 | .43516 | 40 | 13.93 | 190.7 | .43196 | 40 | 10.44 | 190.5 | .42256 |
| 20 | 20.14 | 187.7 | .43847 | | | | | | | | |
| 30 | 20.57 | 189.3 | .44161 | **50** | 14.23 | 192.3 | .43517 | **50** | 10.66 | 192.2 | .42582 |
| 40 | 21.00 | 190.9 | .44491 | 60 | 14.52 | 193.9 | .43833 | 60 | 10.88 | 193.8 | .42903 |
| | | | | 70 | 14.71 | 195.6 | .44140 | 70 | 11.10 | 195.5 | .43216 |
| **50** | 21.42 | 192.5 | .44806 | 80 | 15.11 | 197.2 | .44443 | 80 | 11.32 | 197.1 | .43524 |
| 60 | 21.85 | 194.1 | .45116 | 90 | 15.40 | 199.9 | .44741 | 90 | 11.54 | 198.8 | .43825 |
| 70 | 22.27 | 196.7 | .45421 | | | | | | | | |
| 80 | 22.70 | 197.3 | .45722 | **100** | 15.69 | 200.5 | .45035 | **100** | 11.75 | 200.4 | .44123 |
| 90 | 23.12 | 198.9 | .46018 | 110 | 16.97 | 202.2 | .45326 | 110 | 11.97 | 202.1 | .44416 |
| | | | | 120 | 16.26 | 203.8 | .45613 | 120 | 12.18 | 203.7 | .44705 |
| **100** | 23.54 | 200.5 | .46311 | 130 | 16.54 | 205.3 | .45890 | 130 | 12.39 | 205.4 | .44990 |
| 110 | 23.96 | 202.1 | .46600 | 140 | 16.82 | 207.1 | .46176 | 140 | 12.61 | 207.0 | .45271 |
| 120 | 24.39 | 203.8 | .46885 | | | | | | | | |
| 130 | 24.81 | 205.2 | .47167 | **150** | 17.09 | 208.8 | .46451 | **150** | 12.82 | 208.8 | .45543 |
| 140 | 25.23 | 207.1 | .47445 | 160 | 17.35 | 210.4 | .46722 | 160 | 13.03 | 210.3 | .45820 |
| | | | | 170 | 17.62 | 212.1 | .46990 | 170 | 13.24 | 212.0 | .46089 |
| **150** | 25.65 | 208.8 | .47720 | 180 | 17.88 | 213.7 | .47254 | 180 | 13.46 | 213.6 | .46353 |
| 160 | 26.08 | 210.4 | .47991 | 190 | 18.13 | 215.4 | .47514 | 190 | 13.66 | 215.3 | .46614 |
| 170 | 26.50 | 212.1 | .48259 | | | | | | | | |
| 180 | 26.92 | 213.8 | .48523 | **200** | 18.38 | 217.0 | .47769 | **200** | 13.88 | 216.9 | .46871 |

TABLE 56b—*Continued*

| Temperature, Degrees F. | 10 *−1.34 Degrees* | | | Temperature, Degrees F. | 15 *14.43 Degrees* | | | Temperature, Degrees F. | 20 *26.44 Degrees* | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | *v* | *i* | *s* | | *v* | *i* | *s* | | *v* | *i* | *s* |
| *Saturation* | *7.520* | *182.95* | *.40000* | *Saturation* | *5.110* | *184.17* | *.39091* | *Saturation* | *3.878* | *184.86* | *.38329* |
| **0** | 7.545 | 183.2 | .40046 | 20 | 5.192 | 185.4 | .39270 | 40 | 4.035 | 187.8 | .38959 |
| 10 | 7.744 | 185.0 | .40432 | 30 | 5.333 | 187.3 | .39672 | | | | |
| 20 | 7.939 | 186.7 | .40802 | 40 | 5.470 | 189.2 | .40054 | **50** | 4.145 | 189.8 | .39346 |
| 30 | 8.030 | 188.4 | .41159 | | | | | 60 | 4.251 | 191.8 | .39719 |
| 40 | 8.316 | 190.1 | .41505 | **50** | 5.604 | 191.0 | .40424 | 70 | 4.354 | 193.7 | .40080 |
| | | | | 60 | 5.734 | 192.8 | .40777 | 80 | 4.454 | 195.6 | .40429 |
| **50** | 8.500 | 191.8 | .41837 | 70 | 5.862 | 195.6 | .41116 | 90 | 4.552 | 197.5 | .40758 |
| 60 | 8.681 | 193.5 | .42161 | 80 | 5.988 | 196.4 | .41443 | | | | |
| 70 | 8.860 | 195.2 | .42480 | 90 | 6.112 | 198.2 | .41765 | **100** | 4.648 | 199.3 | .41093 |
| 80 | 9.038 | 196.9 | .42795 | | | | | 110 | 4.742 | 201.1 | .41415 |
| 90 | 9.214 | 198.6 | .43104 | **100** | 6.233 | 199.9 | .42076 | 120 | 4.834 | 202.9 | .41726 |
| | | | | 110 | 6.353 | 201.6 | .42383 | 130 | 4.925 | 204.7 | .42027 |
| **100** | 9.389 | 200.3 | .43407 | 120 | 6.471 | 203.3 | .42682 | 140 | 5.015 | 206.5 | .42322 |
| 110 | 9.563 | 202.0 | .43705 | 130 | 6.588 | 205.6 | .48976 | | | | |
| 120 | 9.736 | 203.7 | .43997 | 140 | 6.705 | 206.7 | .43264 | **150** | 5.104 | 208.2 | .42613 |
| 130 | 9.908 | 205.4 | .44283 | | | | | 160 | 5.193 | 209.9 | .42898 |
| 140 | 10.08 | 207.1 | .44565 | **150** | 6.821 | 208.4 | .43548 | 170 | 5.281 | 211.6 | .43176 |
| | | | | 160 | 6.937 | 210.1 | .43825 | 180 | 5.369 | 213.3 | .43449 |
| **150** | 10.25 | 208.8 | .44842 | 170 | 7.052 | 211.8 | .44097 | 190 | 5.456 | 215.0 | .43716 |
| 160 | 10.42 | 210.5 | .45116 | 180 | 7.167 | 213.5 | .44366 | | | | |
| 170 | 10.59 | 212.2 | .45296 | 190 | 7.282 | 215.2 | .44630 | **200** | 5.542 | 216.7 | .43977 |
| 180 | 10.76 | 213.8 | .45651 | | | | | 210 | 5.629 | 218.4 | .44234 |
| 190 | 10.93 | 215.4 | .45913 | **200** | 7.396 | 216.9 | .44889 | 220 | 5.715 | 220.1 | .44488 |
| **200** | 11.10 | 217.0 | .46171 | | | | | | | | |

| Temperature, Degrees F. | 25 *36.33 Degrees* | | | Temperature, Degrees F. | 30 *44.76 Degrees* | | | Temperature, Degrees F. | 40 *58.83 Degrees* | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | *v* | *i* | *s* | | *v* | *i* | *s* | | *v* | *i* | *s* |
| *Saturation* | *3.123* | *185.26* | *.37754* | *Saturation* | *2.614* | *185.48* | *.37269* | *Saturation* | *1.970* | *185.60* | *.36470* |
| 40 | 3.181 | 186.1 | .37927 | 60 | 2.747 | 189.3 | .37969 | 60 | 1.980 | 185.9 | .3654 |
| | | | | 70 | 2.830 | 191.6 | .38428 | 70 | 2.064 | 188.7 | .37064 |
| **50** | 3.273 | 188.4 | .38372 | 80 | 2.907 | 193.8 | .38848 | 80 | 2.121 | 191.3 | .37544 |
| 60 | 3.363 | 190.6 | .38795 | 90 | 2.980 | 195.9 | .39236 | 90 | 2.185 | 193.6 | .37992 |
| 70 | 3.451 | 192.7 | .39198 | | | | | | | | |
| 80 | 3.536 | 194.7 | .39582 | **100** | 3.052 | 197.9 | .39603 | **100** | 2.246 | 196.1 | .38415 |
| 90 | 3.618 | 196.7 | .39945 | 110 | 3.122 | 199.9 | .39955 | 110 | 2.304 | 198.3 | .38810 |
| | | | | 120 | 3.189 | 201.8 | .40293 | 120 | 2.360 | 200.4 | .39183 |
| **100** | 3.696 | 198.6 | .40291 | 130 | 3.254 | 203.7 | .40319 | 130 | 2.413 | 202.5 | .39541 |
| 110 | 3.772 | 200.5 | .40625 | 140 | 3.318 | 205.6 | .40935 | 140 | 2.465 | 204.6 | .39881 |
| 120 | 3.848 | 202.4 | .40949 | | | | | | | | |
| 130 | 3.923 | 204.2 | .41261 | **150** | 3.381 | 207.5 | .41241 | **150** | 2.515 | 206.5 | .40209 |
| 140 | 3.998 | 206.0 | .41568 | 160 | 3.443 | 209.3 | .41539 | 160 | 2.565 | 208.5 | .41525 |
| | | | | 170 | 3.501 | 211.1 | .41829 | 170 | 2.614 | 210.4 | .41831 |
| **150** | 4.073 | 207.8 | .41866 | 180 | 3.565 | 212.9 | .42112 | 180 | 2.662 | 212.3 | .41127 |
| 160 | 4.115 | 209.6 | .42156 | 190 | 3.625 | 214.7 | .42387 | 190 | 2.709 | 214.2 | .41416 |
| 170 | 4.216 | 211.4 | .42439 | | | | | | | | |
| 180 | 4.287 | 213.2 | .43717 | **200** | 3.685 | 216.5 | .42657 | **200** | 2.755 | 216.0 | .41694 |
| 190 | 4.358 | 215.0 | .42988 | 210 | 3.744 | 218.3 | .42921 | 210 | 2.800 | 217.9 | .41966 |
| | | | | 220 | 3.803 | 220.1 | .43180 | 220 | 2.845 | 219.7 | .42233 |
| **200** | 4.428 | 216.7 | .43253 | 230 | 3.861 | 221.9 | .43438 | 230 | 2.889 | 221.5 | .42494 |
| 210 | 4.498 | 218.4 | .43413 | 240 | 3.919 | 223.6 | .43691 | 240 | 2.933 | 223.3 | .42751 |
| 220 | 4.567 | 220.1 | .43769 | | | | | | | | |
| 230 | 4.637 | 221.8 | .44023 | **250** | 3.977 | 225.3 | .43942 | 250 | 2.977 | 225.1 | .43007 |
| 240 | 4.706 | 223.5 | .44275 | 260 | 4.035 | 227.0 | .44188 | 260 | 3.021 | 227.0 | .43262 |

TABLE 56b—*Continued*

| Temperature, Degrees F. | 50 *70.40 Degrees* | | | Temperature, Degrees F. | 60 *80.29 Degrees* | | | Temperature, Degrees F. | 70 *88.97 Degrees* | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | *v* | *i* | *s* | | *v* | *i* | *s* | | *v* | *i* | *s* |
| *Saturation* | *1.577* | *185.45* | *.35826* | *Saturation* | *1.3144* | *186.16* | *.35272* | *Saturation* | *1.125* | *184.77* | *.34789* |
| 80 | 1.668 | 188.4 | .36366 | **100** | 1.288 | 191.4 | .36403 | **100** | 1.181 | 187.6 | .35443 |
| 90 | 1.723 | 191.2 | .36887 | 110 | 1.346 | 194.3 | .36906 | 110 | 1.228 | 191.6 | .36020 |
| | | | | 120 | 1.403 | 197.0 | .37375 | 120 | 1.272 | 194.8 | .36545 |
| **100** | 1.775 | 193.9 | .37369 | 130 | 1.459 | 199.5 | .37810 | 130 | 1.313 | 197.6 | .37028 |
| 110 | 1.825 | 196.4 | .37815 | 140 | 1.514 | 201.9 | .38217 | 140 | 1.352 | 200.3 | .37478 |
| 120 | 1.872 | 198.8 | .38234 | | | | | | | | |
| 130 | 1.917 | 201.1 | .38627 | **150** | 1.563 | 204.2 | .38603 | **150** | 1.389 | 202.9 | .37897 |
| 140 | 1.961 | 203.3 | .38998 | 160 | 1.608 | 206.5 | .38963 | 160 | 1.424 | 205.3 | .38291 |
| | | | | 170 | 1.650 | 208.6 | .39310 | 170 | 1.457 | 207.6 | .38662 |
| **150** | 2.003 | 205.4 | .39353 | 180 | 1.689 | 210.7 | .39639 | 180 | 1.489 | 209.9 | .39014 |
| 160 | 2.044 | 207.5 | .39691 | 190 | 1.726 | 212.8 | .39956 | 190 | 1.521 | 212.0 | .39348 |
| 170 | 2.084 | 209.6 | .40015 | | | | | | | | |
| 180 | 2.123 | 211.6 | .40327 | **200** | 1.751 | 214.8 | .40260 | **200** | 1.551 | 214.1 | .39670 |
| 190 | 2.161 | 213.4 | .40628 | 210 | 1.785 | 216.8 | .40554 | 210 | 1.580 | 216.1 | .39978 |
| | | | | 220 | 1.819 | 218.7 | .40839 | 220 | 1.608 | 218.1 | .40275 |
| **200** | 2.199 | 215.4 | .40919 | 230 | 1.853 | 220.7 | .41118 | 230 | 1.636 | 200.1 | .40564 |
| 210 | 2.237 | 217.3 | .41200 | 240 | 1.885 | 222.6 | .41391 | 240 | 1.664 | 222.1 | .40845 |
| 220 | 2.274 | 219.2 | .41477 | | | | | | | | |
| 230 | 2.311 | 221.1 | .41748 | **250** | 1.917 | 224.5 | .41657 | **250** | 1.691 | 224.1 | .41120 |
| 240 | 2.347 | 223.0 | .42015 | 260 | 1.948 | 226.4 | .41917 | 260 | 1.718 | 226.0 | .41389 |
| | | | | 270 | 1.979 | 228.2 | .42175 | 270 | 1.745 | 227.9 | .41653 |
| **250** | 2.383 | 224.9 | .42275 | 280 | 2.010 | 230.1 | .42431 | 280 | 1.771 | 229.8 | .41912 |
| 260 | 2.418 | 226.7 | .42535 | 290 | 2.040 | 232.0 | .42685 | 290 | 1.798 | 231.7 | .42167 |
| 270 | 2.454 | 228.5 | .42791 | | | | | | | | |
| 280 | 2.489 | 230.3 | .43045 | **300** | 2.070 | 233.8 | .42935 | **300** | 1.824 | 233.5 | .42418 |

| Temperature, Degrees F. | 80 *96.88 Degrees* | | | Temperature, Degrees F. | 100 *110.15 Degrees* | | | Temperature, Degrees F. | 120 *121.52 Degrees* | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | *v* | *i* | *s* | | *v* | *i* | *s* | | *v* | *i* | *s* |
| *Saturation* | *.9809* | *184.33* | *.34357* | *Saturation* | *.7786* | *183.30* | *.33603* | *Saturation* | *.6430* | *182.19* | *.33954* |
| **100** | .993 | 185.6 | .34571 | 120 | .8190 | 187.3 | .34296 | 140 | .7085 | 190.1 | .34264 |
| 110 | 1.040 | 189.1 | .35214 | 130 | .8575 | 191.0 | .34942 | | | | |
| 120 | 1.084 | 192.5 | .35797 | 140 | .8928 | 194.6 | .35528 | **150** | .7403 | 193.9 | .34904 |
| 130 | 1.125 | 195.7 | .36330 | | | | | 160 | .7700 | 197.4 | .35484 |
| 140 | 1.163 | 198.6 | .36819 | **150** | .9255 | 197.9 | .36061 | 170 | .7972 | 200.6 | .36012 |
| | | | | 160 | .9561 | 200.9 | .36558 | 180 | .8228 | 203.7 | .36494 |
| **150** | 1.199 | 201.3 | .37270 | 170 | .9848 | 203.7 | .37009 | 190 | .8470 | 206.7 | .36936 |
| 160 | 1.232 | 203.9 | .37692 | 180 | 1.012 | 206.4 | .37431 | | | | |
| 170 | 1.263 | 206.4 | .38093 | 190 | 1.038 | 209.0 | .37829 | 200 | .8699 | 209.4 | .37348 |
| 180 | 1.292 | 208.7 | .38461 | | | | | 210 | .8916 | 212.0 | .37737 |
| 190 | 1.370 | 211.0 | .38813 | **200** | 1.062 | 211.5 | .38203 | 220 | .9124 | 214.5 | .38104 |
| | | | | 210 | 1.086 | 213.8 | .38556 | 230 | .9324 | 217.0 | 38451 |
| **200** | 1.347 | 213.3 | .39150 | 220 | 1.109 | 216.1 | .38892 | 240 | .9515 | 219.3 | .38785 |
| 210 | 1.374 | 215.5 | .39471 | 230 | 1.131 | 218.4 | .39214 | | | | |
| 220 | 1.400 | 217.5 | .39780 | 240 | 1.152 | 220.5 | .39524 | **250** | .9700 | 221.5 | .39106 |
| 230 | 1.426 | 219.6 | .40079 | | | | | 260 | .9880 | 223.7 | .39416 |
| 240 | 1.451 | 221.6 | .40369 | **250** | 1.173 | 222.6 | .39824 | 270 | 1.006 | 225.9 | .39713 |
| | | | | 260 | 1.194 | 224.7 | .40114 | 280 | 1.023 | 228.0 | .40002 |
| **250** | 1.476 | 223.6 | .40651 | 270 | 1.213 | 226.8 | .40397 | 290 | 1.040 | 230.1 | .40284 |
| 260 | 1.500 | 225.6 | .40926 | 280 | 1.232 | 228.8 | .40673 | | | | |
| 270 | 1.524 | 227.6 | .41195 | 290 | 1.251 | 230.8 | .40944 | **300** | 1.056 | 233.2 | .40558 |
| 280 | 1.547 | 229.5 | .41459 | | | | | 310 | 1.072 | 234.3 | .40825 |
| 290 | 1.570 | 231.5 | .41719 | **300** | 1.268 | 232.8 | .41207 | 320 | 1.088 | 236.3 | .41085 |
| | | | | 310 | 1.284 | 234.8 | .41464 | 330 | 1.104 | 238.3 | .41338 |
| **300** | 1.593 | 233.4 | .41974 | 320 | 1.299 | 236.7 | .41716 | 340 | 1.120 | 240.3 | .41583 |

| Temperature, Degrees F. | 140 *131.64 Degrees* | | | Temperature, Degrees F. | 140 *131.64 Degrees* | | | Temperature, Degrees F. | 140 *131.64 Degrees* | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | *v* | *i* | *s* | | *v* | *i* | *s* | | *v* | *i* | *s* |
| *Saturation* | *.5451* | *181.04* | *.32388* | *Saturation* | *.5451* | *181.04* | *.32388* | *Saturation* | *.5451* | *181.04* | *.32388* |
| 140 | .5734 | 185.1 | .33089 | 210 | .7513 | 210.0 | .36976 | 280 | .8720 | 227.1 | .39408 |
| **150** | .6055 | 189.7 | .33777 | 220 | .7707 | 212.7 | .37379 | 290 | .8970 | 229.3 | .39701 |
| 160 | .6345 | 193.6 | .34442 | 230 | .7892 | 215.4 | .37758 | **300** | .9017 | 231.5 | .39985 |
| 170 | .6613 | 196.3 | .35041 | 240 | .8070 | 217.9 | .38118 | 310 | .9161 | 233.6 | .40261 |
| 180 | .6861 | 200.8 | .35588 | **250** | .8241 | 220.3 | .38461 | 320 | .9302 | 235.7 | .40529 |
| 190 | .7092 | 204.0 | .36088 | 260 | .8405 | 222.6 | .38789 | 330 | .9441 | 237.7 | .40791 |
| **200** | .7309 | 207.1 | .36548 | 270 | .8564 | 224.9 | .39105 | 340 | .9579 | 239.7 | .41049 |

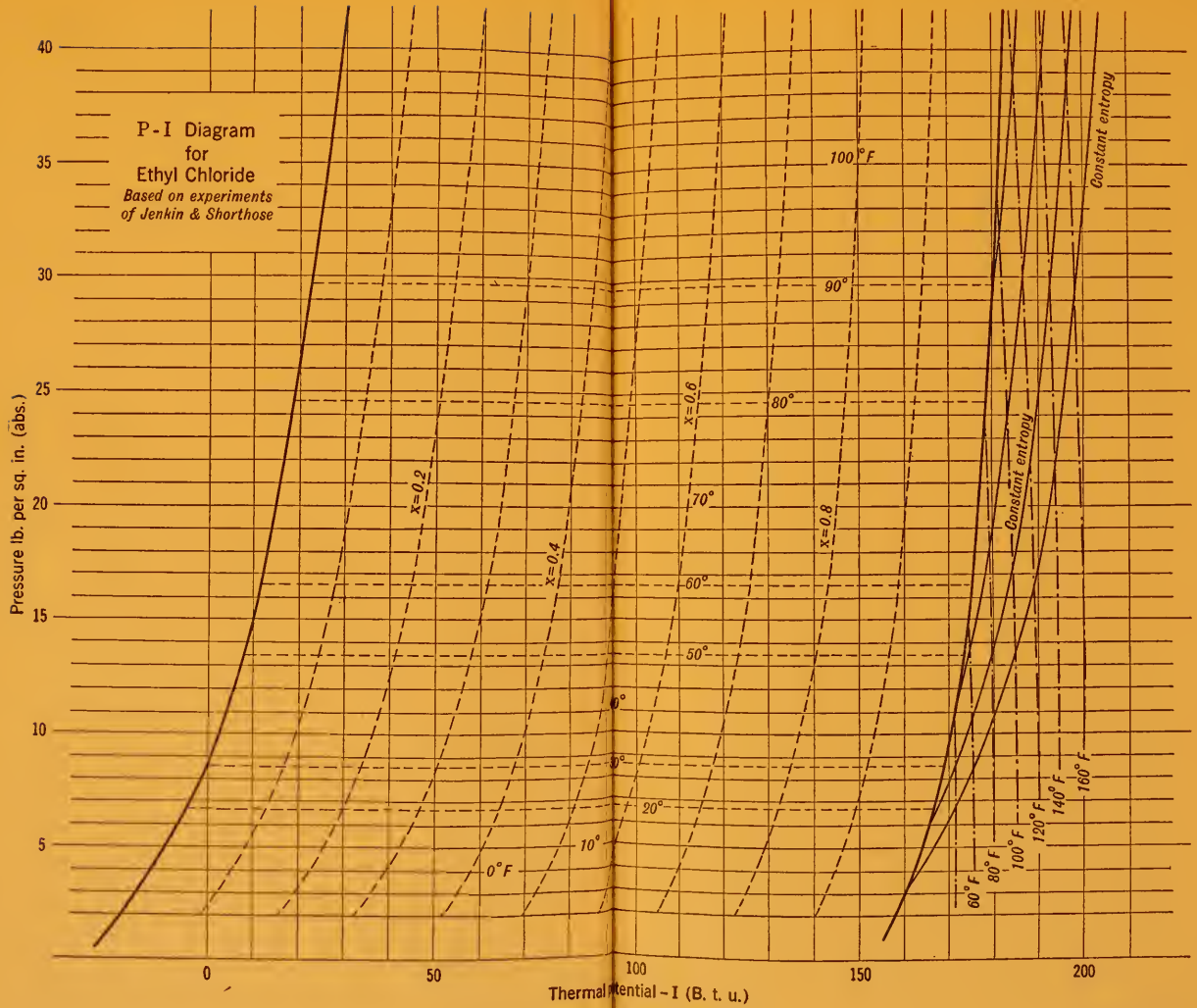



FIG. 167.—P-I Diagram for Ethyl Chloride.

*To face page 241.*

## TABLE 57

### Properties of Ethyl Chloride

| Temperature, Degrees F. | Pressure, Pounds per Square Inch, Absolute | Weight, 1 Cubic Foot of Liquid | Specific Volume of Saturated Gas, Cubic Feet | Thermal Potential of the Liquid, B.t.u. | Latent Heat of Vaporization, B.t.u. | Thermal Potential of Saturated Vapor, B.t.u. |
|---|---|---|---|---|---|---|
| −24 | 1.90 | 60.40 | 32.85 | −20.4 | 178.2 | 157.8 |
| −22 | 2.02 | 60.30 | 31.13 | −19.6 | 177.9 | 158.3 |
| −20 | 2.16 | 60.20 | 29.54 | −19.0 | 177.6 | 158.6 |
| −18 | 2.31 | 60.10 | 28.12 | −18.2 | 177.2 | 159.0 |
| −16 | 2.45 | 60.01 | 26.73 | −17.5 | 176.9 | 159.4 |
| −14 | 2.61 | 59.93 | 25.41 | −16.8 | 176.6 | 159.8 |
| −12 | 2.77 | 59.85 | 24.14 | −16.1 | 176.3 | 160.2 |
| −10 | 2.94 | 59.78 | 22.95 | −15.4 | 175.9 | 160.5 |
| − 8 | 3.13 | 59.70 | 21.84 | −14.6 | 175.6 | 161.0 |
| − 6 | 3.31 | 59.60 | 20.76 | −13.9 | 175.3 | 161.4 |
| − 4 | 3.51 | 59.50 | 19.80 | −13.1 | 174.9 | 161.8 |
| − 2 | 3.71 | 59.40 | 18.89 | −12.4 | 174.6 | 162.2 |
| − 0 | 3.93 | 59.30 | 18.04 | −11.7 | 174.2 | 162.5 |
| + 2 | 4.15 | 59.20 | 17.22 | −11.0 | 173.9 | 162.9 |
| + 4 | 4.38 | 59.10 | 16.46 | −10.2 | 173.6 | 163.4 |
| 6 | 4.61 | 58.99 | 15.74 | − 9.5 | 173.2 | 163.7 |
| 8 | 4.86 | 58.90 | 15.03 | − 8.8 | 172.9 | 164.1 |
| 10 | 5.13 | 58.79 | 14.36 | − 8.1 | 172.6 | 164.5 |
| 12 | 5.42 | 58.69 | 13.74 | − 7.3 | 172.2 | 164.9 |
| 14 | 5.71 | 58.59 | 13.14 | − 6.6 | 171.9 | 165.3 |
| 16 | 6.01 | 58.48 | 12.58 | − 5.9 | 171.6 | 165.7 |
| 18 | 6.32 | 58.39 | 12.06 | − 5.1 | 171.3 | 166.2 |
| 20 | 6.65 | 58.29 | 11.56 | − 4.4 | 170.9 | 166.5 |
| 22 | 6.98 | 58.20 | 11.07 | − 3.6 | 170.6 | 167.0 |
| 24 | 7.31 | 58.10 | 10.58 | − 2.9 | 170.2 | 167.3 |
| 26 | 7.65 | 58.00 | 10.11 | − 2.1 | 169.9 | 167.8 |
| 28 | 8.01 | 57.89 | 9.66 | − 1.4 | 169.6 | 168.2 |
| 30 | 8.40 | 57.78 | 9.22 | − 0.6 | 169.2 | 168.6 |
| 32 | 8.80 | 57.66 | 8.81 | + 0.1 | 168.9 | 169.0 |
| 34 | 9.21 | 57.55 | 8.44 | 0.9 | 168.5 | 169.4 |
| 36 | 9.63 | 57.45 | 8.10 | 1.6 | 168.2 | 169.8 |
| 38 | 10.06 | 57.34 | 7.76 | 2.4 | 167.8 | 170.2 |
| 40 | 10.53 | 57.24 | 7.43 | 3.2 | 167.5 | 170.7 |
| 42 | 11.01 | 57.14 | 7.12 | 3.9 | 167.2 | 171.1 |
| 44 | 11.52 | 57.04 | 6.84 | 4.7 | 166.8 | 171.5 |
| 46 | 12.02 | 56.93 | 6.56 | 5.5 | 166.4 | 171.9 |
| 48 | 12.60 | 56.83 | 6.30 | 6.2 | 166.0 | 172.2 |
| 50 | 13.20 | 56.72 | 6.04 | 6.9 | 165.6 | 172.5 |
| 52 | 13.81 | 56.61 | 5.79 | 7.7 | 165.2 | 172.9 |
| 54 | 14.44 | 56.51 | 5.54 | 8.5 | 164.8 | 173.3 |
| 56 | 15.11 | 56.40 | 5.30 | 9.3 | 164.4 | 173.7 |
| 58 | 15.78 | 56.29 | 5.07 | 10.0 | 164.0 | 174.0 |
| 60 | 16.45 | 56.19 | 4.84 | 10.8 | 163.6 | 174.4 |
| 62 | 17.14 | 56.10 | 4.64 | 11.6 | 163.2 | 174.8 |
| 64 | 17.82 | 56.00 | 4.46 | 12.4 | 162.8 | 175.2 |
| 66 | 18.52 | 55.91 | 4.30 | 13.1 | 162.4 | 175.5 |
| 68 | 19.25 | 55.80 | 4.15 | 13.9 | 162.0 | 175.9 |
| 70 | 21.03 | 55.69 | 4.00 | 14.7 | 161.6 | 176.3 |
| 72 | 21.83 | 55.58 | 3.86 | 15.4 | 161.2 | 176.8 |
| 74 | 21.66 | 55.48 | 3.70 | 16.2 | 160.8 | 177.0 |
| 76 | 22.55 | 55.38 | 3.60 | 17.0 | 160.4 | 177.4 |
| 78 | 23.42 | 55.26 | 3.49 | 17.8 | 160.0 | 177.8 |
| 80 | 24.33 | 55.15 | 3.39 | 18.6 | 159.5 | 178.1 |
| 82 | 25.25 | 55.04 | 3.30 | 19.4 | 159.1 | 178.5 |
| 84 | 26.19 | 54.94 | 3.21 | 20.2 | 158.7 | 178.9 |
| 86 | 27.15 | 54.83 | 3.13 | 21.0 | 158.2 | 179.2 |
| 88 | 28.17 | 54.72 | 3.06 | 21.7 | 157.8 | 179.5 |
| 90 | 29.24 | 54.61 | 2.99 | 22.6 | 157.3 | 179.9 |
| 92 | 30.33 | 54.50 | 2.92 | 23.4 | 156.8 | 180.2 |
| 94 | 31.45 | 54.40 | 2.86 | 24.1 | 156.3 | 180.4 |
| 96 | 32.60 | 54.30 | 2.80 | 25.0 | 155.8 | 180.8 |
| 98 | 33.75 | 54.20 | 2.75 | 25.8 | 155.4 | 181.2 |
| 100 | 34.93 | 54.10 | 2.70 | 26.6 | 154.9 | 181.5 |
| 102 | 36.14 | 54.00 | 2.66 | 27.4 | 154.5 | 181.9 |
| 104 | 37.36 | 53.90 | 2.63 | 28.2 | 154.0 | 182.2 |
| 106 | 38.59 | 53.79 | 2.58 | 29.0 | 153.6 | 182.6 |
| 108 | 39.82 | 53.68 | 2.54 | 29.8 | 153.1 | 182.9 |
| 110 | 41.10 | 53.56 | 2.51 | 30.5 | 152.6 | 183.1 |

## TABLE 58

### Thermodynamic Properties of Methyl Chloride ($CH_3CL$)

| | Temperature, Degrees F. | Pressure (Absolute) | Pressure (Gage) | Specific Volume of Saturated Gas, Cubic Feet per Pound | Density of Liquid, Pounds per Cubic Foot | Total Heat, Liquid, B.t.u. per Pound | Latent Heat, B.t.u. per Pound | Heat of Vapor, B.t.u. per Pound |
|---|---|---|---|---|---|---|---|---|
| | −24 | 10.60 | 8.4* | 8.83 | 63.435 | −20.45 | 187.22 | 166.77 |
| † | −22 | 11.20 | 7.4* | 8.466 | 63.310 | −19.68 | 186.80 | 167.12 |
| | −20 | 11.75 | 6.1* | 8.09 | 63.185 | −19.0 | 186.36 | 167.36 |
| | −18 | 12.50 | 4.8* | 7.73 | 63.060 | −18.25 | 185.92 | 167.67 |
| | −16 | 13.1 | 3.2* | 7.38 | 62.935 | −17.57 | 185.50 | 167.93 |
| | −14 | 13.75 | 1.4* | 7.06 | 62.180 | −16.81 | 185.07 | 168.26 |
| | −12 | 14.4 | 0.8* | 6.75 | 62.685 | −16.10 | 184.64 | 168.54 |
| | −10 | 15.0 | 0.3 | 6.46 | 62.560 | −15.38 | 184.21 | 168.83 |
| | − 8 | 15.8 | 1.1 | 6.18 | 62.435 | −14.68 | 183.78 | 169.10 |
| | − 6 | 16.5 | 1.8 | 5.92 | 62.310 | −13.95 | 183.33 | 169.38 |
| † | − 4 | 17.08 | 2.3 | 5.67 | 62.185 | −13.20 | 182.90 | 169.70 |
| | − 2 | 17.9 | 3.2 | 5.41 | 62.061 | −12.50 | 182.44 | 169.94 |
| | 0 | 18.8 | 4.1 | 5.18 | 61.936 | −11.75 | 181.98 | 170.23 |
| | 2 | 19.6 | 4.9 | 4.96 | 61.811 | −11.0 | 181.52 | 170.52 |
| | 4 | 20.5 | 5.8 | 4.75 | 61.686 | −10.30 | 181.08 | 170.78 |
| | 6 | 21.5 | 6.8 | 4.55 | 61.561 | − 9.55 | 180.60 | 171.05 |
| | 8 | 22.4 | 7.7 | 4.36 | 61.436 | − 8.80 | 180.12 | 171.32 |
| | 10 | 23.3 | 8.6 | 4.18 | 61.311 | − 8.06 | 179.65 | 171.59 |
| | 12 | 24.4 | 9.7 | 4.02 | 61.187 | − 7.30 | 179.18 | 171.88 |
| † | 14 | 25.34 | 10.6 | 3.86 | 61.086 | − 6.62 | 178.70 | 172.08 |
| | 16 | 26.5 | 11.8 | 3.70 | 60.959 | − 5.85 | 178.23 | 172.40 |
| | 18 | 27.6 | 12.9 | 3.55 | 60.831 | − 5.10 | 177.75 | 172.65 |
| | 20 | 28.8 | 14.1 | 3.41 | 60.702 | − 4.32 | 177.27 | 172.95 |
| | 22 | 29.8 | 15.1 | 3.28 | 60.593 | − 3.60 | 176.80 | 173.20 |
| | 24 | 31.2 | 16.5 | 3.15 | 60.464 | − 2.85 | 176.34 | 173.49 |
| | 26 | 32.5 | 17.8 | 3.03 | 60.365 | − 2.10 | 175.87 | 173.77 |
| | 28 | 33.8 | 19.1 | 2.92 | 60.206 | − 1.38 | 175.38 | 174.00 |
| | 30 | 35.2 | 20.5 | 2.81 | 60.077 | − 0.62 | 174.90 | 174.28 |
| † | 32 | 36.57 | 21.8 | 2.69 | 59.914 | + 0.11 | 174.40 | 174.51 |
| | 34 | 37.9 | 23.2 | 2.59 | 59.779 | 0.87 | 173.92 | 174.79 |
| | 36 | 39.5 | 24.8 | 2.49 | 59.650 | 1.63 | 173.41 | 175.04 |
| | 38 | 41.1 | 26.4 | 2.40 | 59.521 | 2.40 | 172.91 | 175.31 |
| | 40 | 42.6 | 27.9 | 2.31 | 59.492 | 3.15 | 172.42 | 175.57 |
| | 42 | 44.3 | 29.6 | 2.225 | 59.263 | 3.90 | 171.92 | 175.82 |
| | 44 | 46.1 | 31.4 | 2.14 | 59.134 | 4.66 | 171.42 | 176.08 |

| | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | 46 | 47.8 | 33.1 | 2.06 | 59.005 | 5.41 | 170.90 | 176.31 |
| | 48 | 49.6 | 34.9 | 1.99 | 58.876 | 6.19 | 170.40 | 176.59 |
| † | 50 | 51.51 | 36.77 | 1.93 | 58.747 | 6.88 | 169.90 | 176.78 |
| | 52 | 53.5 | 38.8 | 1.86 | 58.616 | 7.66 | 169.40 | 177.06 |
| | 54 | 55.4 | 40.7 | 1.79 | 58.484 | 8.40 | 168.90 | 177.30 |
| | 56 | 57.4 | 42.7 | 1.72 | 58.353 | 9.20 | 168.40 | 177.60 |
| | 58 | 59.3 | 44.6 | 1.67 | 58.220 | 9.95 | 167.88 | 177.83 |
| | 60 | 61.6 | 46.9 | 1.61 | 58.077 | 10.70 | 167.35 | 178.05 |
| | 62 | 63.8 | 49.1 | 1.55 | 57.943 | 11.48 | 166.83 | 178.31 |
| | 64 | 66.1 | 51.4 | 1.50 | 57.809 | 12.25 | 166.29 | 178.54 |
| | 66 | 68.4 | 53.7 | 1.45 | 57.675 | 13.00 | 165.74 | 178.74 |
| † | 68 | 70.97 | 56.23 | 1.39 | 57.541 | 13.75 | 165.20 | 178.95 |
| | 70 | 73.3 | 58.6 | 1.34 | 57.403 | 14.52 | 164.65 | 179.17 |
| | 72 | 75.8 | 61.1 | 1.30 | 57.265 | 15.30 | 164.10 | 179.40 |
| | 74 | 78.5 | 63.8 | 1.26 | 57.127 | 16.08 | 163.56 | 179.64 |
| | 76 | 79.8 | 65.1 | 1.22 | 56.989 | 16.83 | 163.00 | 179.83 |
| | 78 | 82.5 | 67.8 | 1.17 | 56.851 | 17.59 | 162.44 | 180.03 |
| | 80 | 85.3 | 70.6 | 1.14 | 56.714 | 18.36 | 161.88 | 180.24 |
| | 82 | 89.7 | 75.0 | 1.10 | 56.576 | 19.12 | 161.31 | 180.43 |
| | 84 | 92.7 | 78.0 | 1.06 | 56.438 | 19.88 | 160.75 | 180.63 |
| † | 86 | 95.52 | 80.78 | 1.04 | 56.300 | 20.64 | 160.20 | 180.84 |
| | 88 | 98.9 | 84.2 | 1.00 | 56.161 | 21.34 | 159.65 | 180.99 |
| | 90 | 102.1 | 87.4 | 0.98 | 56.022 | 22.13 | 159.09 | 181.22 |
| | 92 | 105.3 | 90.6 | .95 | 55.883 | 22.90 | 158.52 | 181.42 |
| | 94 | 108.6 | 93.9 | .92 | 55.744 | 23.68 | 157.98 | 181.66 |
| | 96 | 112.0 | 97.3 | .892 | 55.605 | 24.46 | 157.41 | 181.87 |
| | 98 | 115.3 | 100.6 | .88 | 55.466 | 25.27 | 156.86 | 182.13 |
| | 100 | 118.8 | 104.1 | .85 | 55.327 | 26.06 | 156.30 | 182.36 |
| | 102 | 122.5 | 107.8 | .83 | 55.188 | 26.84 | 155.75 | 182.59 |
| † | 104 | 126.2 | 111.5 | .818 | 55.050 | 27.63 | 155.20 | 182.83 |
| | 106 | 130.0 | 115.3 | .79 | 54.911 | 28.41 | 154.62 | 183.02 |
| | 108 | 133.9 | 119.2 | .78 | 54.772 | 29.22 | 154.04 | 183.26 |
| | 110 | 137.6 | 122.9 | .765 | 54.633 | 30.03 | 153 46 | 183.49 |
| | 112 | 141.9 | 127.2 | .745 | 54.494 | 30.84 | 152.90 | 183.74 |
| | 114 | 146.1 | 131.4 | .730 | 54.355 | 31.67 | 152.33 | 184.00 |
| | 116 | 150.5 | 135.8 | .722 | 54.216 | 32.50 | 151.76 | 184.26 |
| | 118 | 155.0 | 140.3 | .712 | 54 076 | 33.33 | 151.19 | 184.52 |
| | 120 | 159.6 | 144.9 | .700 | 53.936 | 34.16 | 150.62 | 184.78 |
| | 122 | 164.4 | 149.7 | .699 | 53.798 | 34.99 | 150 05 | 185.04 |
| | 124 | 169.2 | 154.5 | .689 | 53.664 | 35.82 | 149.58 | 185.40 |
| | 126 | 174.3 | 159.6 | .678 | 53.530 | 36.65 | 149.01 | 185.66 |
| | 128 | 179.2 | 164.5 | .668 | 53.496 | 37.48 | 148.44 | 185.92 |
| | 130 | 184.3 | 169 6 | .659 | 53.362 | 38.31 | 147.87 | 186.18 |

* Inches of Mercury.
† Values indicated by † are from D. N. Shorthose. Other values by H. J. Macintire.

in many cases in which the required piston displacement can be secured with less difficulty than in the case of the reciprocating compressor.

**Methyl Chloride.**—Methyl chloride ($CH_3Cl$) has considerable resemblance to ethyl chloride, but the unit pressures are greater and the piston displacement per ton of refrigeration is less. It is slightly anæsthetic but not unpleasant to inhale except that some persons are nauseated by the gas, and it is inflammable; much less so than ethyl chloride, but more so than ammonia. The latent heat of vaporization ($r$) is about 170 B.t.u. per lb. It does not seem to be corrosive to copper or compositions of copper, nor to iron or steel, the pressures encountered are 95 lb. per sq. in. abs. (80 lb. gage) or less as a rule (Table 58), and the evaporating pressures are greater than one atmosphere if the boiling temperature is higher than −10 deg. F.

Methyl chloride[5] does not appear to cause any damage to goods or commodities. Both methyl and ethyl chloride are nearly as harmless as carbon dioxide, as a relatively large percentage of these gases in the air will not injure the eyes or lungs and seemingly the bad effects are only from the absence of oxygen, as would be the case with carbon dioxide or nitrogen. There does not seem to be any *cumulative* effect, as would be the case with *carbon monoxide*, which, incidentally, may not show the real effect on a person until some time after the exposure. Both methyl and ethyl chloride must be anhydrous, or the water content will freeze at the expansion valve. Mineral oils dissolve readily in methyl and ethyl chloride, but this is not true in the case of sulphur dioxide.

**Propane.**—Of the gasoline compounds, butane ($C_4H_{10}$), propane ($C_3H_8$) and ethane ($C_2H_6$), propane approximates very closely the physical characteristics—as regards pressures and refrigerating ability—of ammonia, whereas ethane is more like carbon dioxide and butane like sulphur dioxide or ethyl chloride. Propane will give a lower refrigerating temperature than ammonia will for the same suction pressure. According to Dr. E. E. Smith, propane up to 50 per cent in the air (by volume) will not cause injury to people. Table 60[6] gives the limits of the explosibility of propane and other refrigerants. Table 59 gives some of the thermodynamic properties of propane.

**Butane.**—As mentioned, butane has the pressures found in sulphur dioxide practice but it is harmless to breathe. Its particular field—up to the present time—is in the household or the $\frac{1}{6}$-ton (ice cream cabinet) machine now being manufactured in considerable numbers. In either

[5] Certain Physical and Chemical Properties of Methyl Chloride, by H. J. Macintire, C. S. Marvel and S. F. Ford, Amer. Soc. of Refrigerating Engineers, 1927.

[6] H. D. Edwards, Amer. Soc. Refrigerating Engineers, 1924.

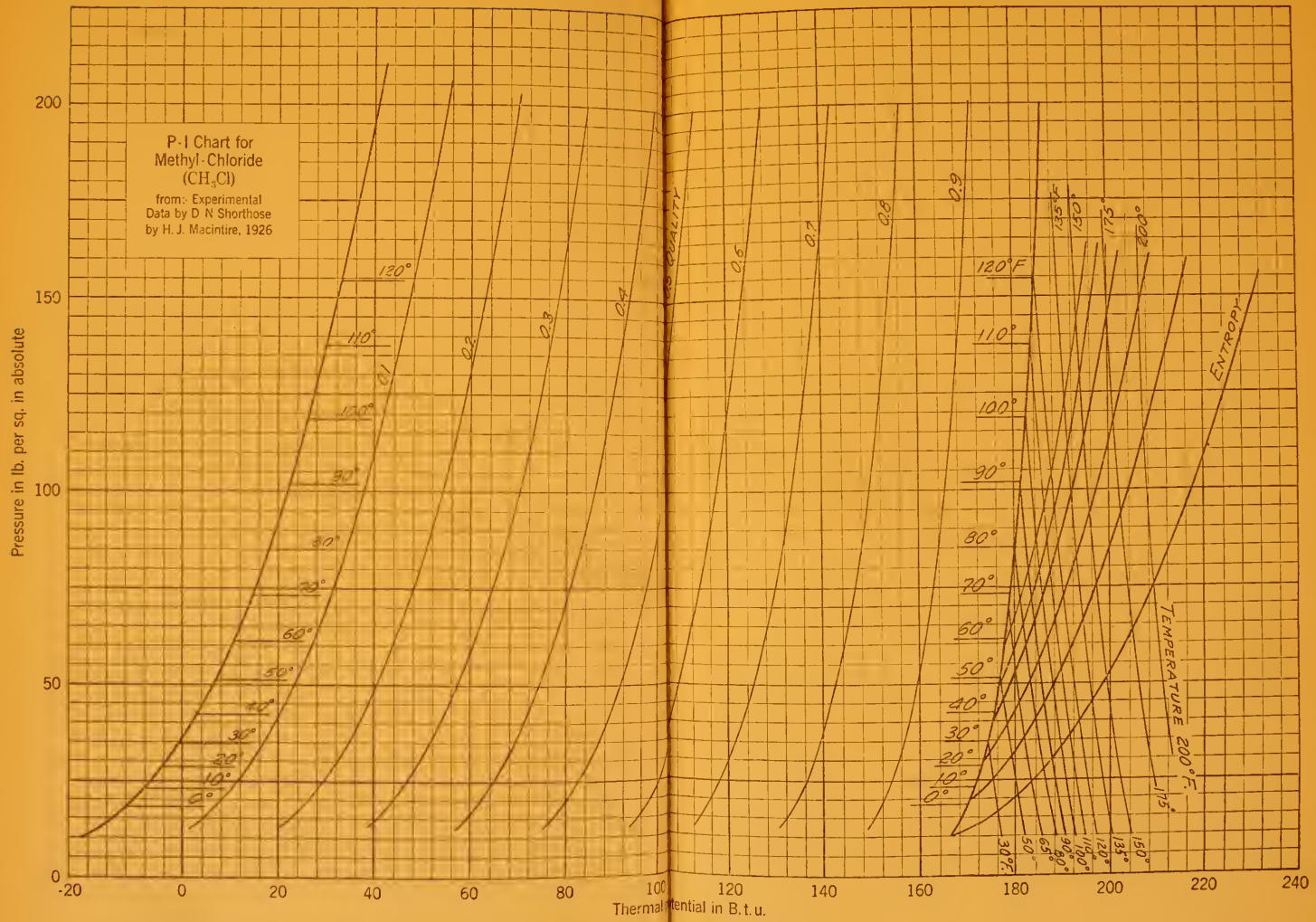



Fig. 168.—P–I Diagram for Methyl Chloride.

*To face page 244.*

## TABLE 59

### Thermodynamic Properties of Propane

| Temperature, Degrees F. $t$ | Pressure, Pounds per Square Inch: Absolute $p$ | Pressure, Pounds per Square Inch: Gage $g.p.$ | Specific Volume, Cubic Feet per Pound: Liquid $v$ | Specific Volume, Cubic Feet per Pound: Vapor $V$ | Density, Pounds per Cubic Foot: Liquid $1/v$ | Density, Pounds per Cubic Foot: Vapor $V$ | Heat Content, B.t.u. per Pound: Liquid $h$ | Heat Content, B.t.u. per Pound: Vapor $H$ | Latent Heat, B.t.u. per Pound $L$ | Entropy, B.t.u. per Pound, Deg. F.: Liquid $s$ | Entropy, B.t.u. per Pound, Deg. F.: Vapor $S$ | Temperature, Degrees F. $t$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| −75 | 6.37 | 17.0* | 0.02660 | 14.5 | 37.59 | 0.0690 | −39.5 | 151.0 | 190.5 | −0.092 | 0.404 | −75 |
| −70 | 7.37 | 14.9* | 0.02674 | 12.9 | 37.40 | 0.0775 | −37.0 | 152.5 | 189.5 | −0.806 | 0.400 | −70 |
| −65 | 8.48 | 12.7* | 0.02688 | 11.3 | 37.20 | 0.0885 | −34.5 | 153.5 | 188.0 | −0.080 | 0.397 | −65 |
| −60 | 9.72 | 10.1* | 0.02703 | 9.93 | 37.00 | 0.111 | −32.0 | 155.0 | 187.0 | −0.074 | 0.393 | −60 |
| −55 | 11.1 | 7.3* | 0.02717 | 8.70 | 36.80 | 0.115 | −29.0 | 156.5 | 185.5 | −0.067 | 0.391 | −55 |
| −50 | 12.6 | 4.3* | 0.02732 | 7.74 | 36.60 | 0.129 | −26.5 | 158.0 | 184.5 | −0.061 | 0.389 | −50 |
| −45 | 14.4 | 0.6* | 0.02748 | 6.89 | 36.39 | 0.145 | −24.0 | 159.0 | 183.0 | −0.055 | 0.386 | −45 |
| −40 | 16.2 | 1.5 | 0.02763 | 6.13 | 36.19 | 0.163 | −21.5 | 160.0 | 181.5 | −0.049 | 0.384 | −40 |
| −35 | 18.1 | 3.4 | 0.02779 | 5.51 | 35.99 | 0.181 | −19.0 | 161.0 | 180.0 | −0.042 | 0.382 | −35 |
| −30 | 20.3 | 5.6 | 0.02795 | 4.93 | 35.78 | 0.203 | −16.0 | 163.0 | 179.0 | −0.036 | 0.380 | −30 |
| −25 | 22.7 | 8.0 | 0.02811 | 4.46 | 35.58 | 0.224 | −13.5 | 164.0 | 177.5 | −0.030 | 0.378 | −25 |
| −20 | 25.4 | 10.7 | 0.02827 | 4.00 | 35.37 | 0.250 | −11.0 | 165.0 | 176.0 | −0.024 | 0.377 | −20 |
| −15 | 28.3 | 13.6 | 0.02844 | 3.60 | 35.16 | 0.278 | − 8.0 | 167.0 | 175.0 | −0.018 | 0.375 | −15 |
| −10 | 31.4 | 16.7 | 0.02860 | 3.26 | 34.96 | 0.307 | − 5.5 | 168.0 | 173.5 | −0.012 | 0.374 | −10 |
| − 5 | 34.7 | 20.0 | 0.02878 | 2.97 | 34.75 | 0.337 | − 2.5 | 169.5 | 172.0 | −0.006 | 0.372 | − 5 |
| 0 | 38.2 | 23.5 | 0.02895 | 2.71 | 34.54 | 0.369 | 0 | 170.5 | 170.5 | 0.000 | 0.371 | 0 |
| + 5 | 41.9 | 27.2 | 0.02913 | 2.48 | 34.33 | 0.403 | + 3.0 | 172.0 | 169.5 | +0.006 | 0.370 | + 5 |
| 10 | 46.0 | 31.3 | 0.02931 | 2.27 | 34.12 | 0.441 | 5.5 | 173.5 | 168.0 | 0.012 | 0.370 | 10 |
| 15 | 50.6 | 35.9 | 0.02950 | 2.07 | 33.90 | 0.483 | 8.5 | 175.0 | 166.5 | 0.018 | 0.369 | 15 |
| 20 | 55.5 | 40.8 | 0.02970 | 1.90 | 33.67 | 0.526 | 11.0 | 176.0 | 165.0 | 0.024 | 0.368 | 20 |
| 25 | 60.9 | 46.2 | 0.02991 | 1.74 | 33.43 | 0.575 | 14.0 | 177.5 | 163.5 | 0.030 | 0.368 | 25 |
| 30 | 66.3 | 51.6 | 0.03012 | 1.60 | 33.20 | 0.625 | 17.0 | 179.0 | 162.0 | 0.035 | 0.366 | 30 |
| 35 | 72.0 | 57.3 | 0.03033 | 1.48 | 32.97 | 0.676 | 20.0 | 180.5 | 160.5 | 0.041 | 0.366 | 35 |
| 40 | 78.0 | 63.3 | 0.03055 | 1.37 | 32.73 | 0.730 | 23.0 | 182.0 | 159.0 | 0.047 | 0.366 | 40 |
| 45 | 84.6 | 69.9 | 0.03078 | 1.27 | 32.49 | 0.787 | 26.0 | 183.5 | 157.5 | 0.053 | 0.365 | 45 |
| 50 | 91.8 | 77.1 | 0.03102 | 1.18 | 32.24 | 0.847 | 29.0 | 185.0 | 156.0 | 0.059 | 0.365 | 50 |
| 55 | 99.3 | 84.6 | 0.03125 | 1.10 | 32.00 | 0.909 | 32.0 | 186.5 | 154.5 | 0.065 | 0.365 | 55 |
| 60 | 107.1 | 92.4 | 0.03150 | 1.01 | 31.75 | 0.990 | 35.0 | 188.0 | 153.0 | 0.070 | 0.364 | 60 |
| 65 | 115.4 | 100.7 | 0.03174 | 0.945 | 31.50 | 1.06 | 38.0 | 189.5 | 151.5 | 0.076 | 0.364 | 65 |
| 70 | 124.0 | 109.3 | 0.03201 | 0.883 | 31.24 | 1.13 | 41.0 | 190.5 | 149.5 | 0.082 | 0.364 | 70 |
| 75 | 133.2 | 118.5 | 0.03229 | 0.825 | 30.97 | 1.21 | 44.0 | 192.0 | 148.0 | 0.088 | 0.364 | 75 |
| 80 | 142.8 | 128.1 | 0.03257 | 0.770 | 30.70 | 1.30 | 47.5 | 193.5 | 146.0 | 0.093 | 0.364 | 80 |
| 85 | 153.1 | 138.4 | 0.03287 | 0.722 | 30.42 | 1.39 | 50.5 | 195.0 | 144.5 | 0.099 | 0.364 | 85 |
| 86 | 155.3 | 140.5 | 0.03292 | 0.717 | 30.38 | 1.40 | 51.0 | 195.0 | 144.0 | 0.100 | 0.364 | 86 |
| 90 | 164.0 | 149.0 | 0.03317 | 0.673 | 30.15 | 1.49 | 54.0 | 196.5 | 142.5 | 0.105 | 0.364 | 90 |
| 95 | 175.0 | 160.0 | 0.03348 | 0.632 | 29.87 | 1.58 | 57.0 | 197.5 | 140.5 | 0.111 | 0.364 | 95 |
| 100 | 187.0 | 172.0 | 0.04481 | 0.591 | 29.58 | 1.69 | 60.5 | 199.0 | 138.5 | 0.116 | 0.363 | 100 |
| 105 | 200.0 | 185.0 | 0.03416 | 0.553 | 29.27 | 1.81 | 63.5 | 200.0 | 136.5 | 0.122 | 0.364 | 105 |
| 100 | 187.0 | 172.0 | 0.03381 | 0.591 | 29.58 | 1.69 | 67.0 | 201.0 | 134.0 | 0.128 | 0.363 | 110 |
| 115 | 226.0 | 211.0 | 0.03493 | 0.488 | 28.63 | 2.05 | 70.5 | 202.0 | 131.5 | 0.134 | 0.363 | 115 |
| 120 | 240.0 | 225.0 | 0.03534 | 0.459 | 28.30 | 2.18 | 73.5 | 202.5 | 129.0 | 0.140 | 0.363 | 120 |
| 125 | 254.0 | 239.0 | 0.03575 | 0.432 | 27.97 | 2.31 | 77.0 | 203.5 | 126.5 | 0.145 | 0.361 | 125 |

* Inches of mercury below one standard atmosphere (29.92 in.).

case medium pressures and safety are the first requirements, and the actual (relative) size is secondary, as is the amount of power required to operate the compressor. Tables 61 and 62 give the thermodynamic properties of butane and isobutane.

TABLE 60

| Refrigerants | Explosion Limits with Air Ratio by Volume | | Maximum Explosion Pressures, Pounds per Square Inch | | Time of Maximum Pressure in Seconds | |
|---|---|---|---|---|---|---|
| | Low | High | *a* | *b* | *a* | *b* |
| Ammonia | 13.1* | 26.8 | 54 | 52 | 0.252 | 0.175 |
| Butane | 1.65 | 5.7 | 102 | 102 | 0.027 | 0.027 |
| Carbon dioxide | | | | | | |
| Ethane | 3.1 | 10.7 | ..... | 108 | ..... | 0.018 |
| Ethyl chloride | 4.3 | 14.0 | 98 | ..... | 0.049 | |
| Gasoline | 1.4 | 6.0 | ..... | 100 | ..... | 0.026 |
| Illuminating gas | 7.0 | 21.0 | ..... | 95 | ..... | 0.017 |
| Methyl chloride | 8.9 | 15.5 | 81 | ..... | 0.099 | |
| Propane | 2.4 | 8.4 | 100 | 104 | 0.023 | 0.020 |
| Sulphur dioxide | | | | | | |

* Lowenstein says the lower value is 19.6. *a* refers to Linde Air Products Laboratories. *b* refers to the Underwriters Laboratory, Chicago.

**Dicloethylene.**—Dicloethylene ($C_2H_2Cl_2$) is being used[7] by only one company in a centrifugal type of rotary compressor. It is a colorless liquid, practically non-explosive and harmless to people and commodities. The unit pressure at the standard condensing temperature of 86 deg. F. is about one-half an atmosphere.

This refrigerant is especially good for centrifugal compression because the molecular weight is high and the unit pressures are very low—being below the atmosphere—although the ratio of the suction to the condensing pressure is nominal. The result is that with *six* stages the actual pressure increase—$6\frac{1}{2}$ lb.—is easily obtained. It will be noted that all pressures are less than that of the atmosphere, and the difficulty, if any, is to remove the air that enters the compressor through the shaft stuffing boxes without excessive loss of the refrigerant. Table 63 which was prepared by Willis Carrier, gives the design requirements with this refrigerant.

[7] W. H. Carrier, American Soc. Refrig. Engineers, June, 1924.

## TABLE 61

### Thermodynamic Properties of Butane ($C_4H_{10}$)

| Temperature, Degrees F. $t$ | Pressure, Pounds per Square Inch | | Specific Volume, Cubic Feet per Pound | | Density, Pounds per Cubic Foot | | Heat Content, B.t.u. per Pound | | Latent Heat, B.t.u. per Pound $L$ | Entropy, B.t.u. per Pound, Deg. F. | | Temperature, Degrees F. $t$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Absolute $p$ | Gage $g.p.$ | Liquid $v$ | Vapor $V$ | Liquid $1/v$ | Vapor $V$ | Liquid $h$ | Vapor $H$ | | Liquid $s$ | Vapor $S$ | |
| 0 | 7.3 | 15.0* | 0.02591 | 11.1 | 38.59 | 0.0901 | 0 | 170.5 | 170.5 | 0.000 | 0.371 | 0 |
| 5 | 8.2 | 13.2* | 0.02603 | 9.98 | 38.41 | 0.100 | 2.5 | 172.0 | 169.5 | 0.006 | 0.370 | 5 |
| 10 | 9.2 | 11.1* | 0.02615 | 8.95 | 38.24 | 0.112 | 5.5 | 174.0 | 168.5 | 0.011 | 0.370 | 10 |
| 15 | 10.4 | 8.8* | 0.02627 | 8.05 | 38.07 | 0.124 | 8.0 | 176.0 | 168.0 | 0.017 | 0.371 | 15 |
| 20 | 11.6 | 6.3* | 0.02639 | 7.23 | 37.89 | 0.138 | 10.5 | 177.5 | 167.0 | 0.022 | 0.370 | 20 |
| 25 | 13.0 | 3.6* | 0.02651 | 6.55 | 37.72 | 0.153 | 13.0 | 179.0 | 166.0 | 0.028 | 0.371 | 25 |
| 30 | 14.4 | 0.6* | 0.02661 | 5.90 | 37.54 | 0.169 | 16.0 | 181.5 | 165.5 | 0.033 | 0.371 | 30 |
| 35 | 16.0 | 1.3 | 0.02676 | 5.37 | 37.37 | 0.186 | 19.0 | 183.5 | 164.5 | 0.039 | 0.371 | 35 |
| 40 | 17.7 | 3.0 | 0.02689 | 4.88 | 37.19 | 0.205 | 21.5 | 185.0 | 163.5 | 0.044 | 0.371 | 40 |
| 45 | 19.6 | 4.9 | 0.02703 | 4.47 | 37.00 | 0.224 | 24.5 | 187.0 | 162.5 | 0.050 | 0.372 | 45 |
| 50 | 21.6 | 6.9 | 0.02716 | 4.07 | 36.82 | 0.246 | 27.0 | 188.5 | 161.5 | 0.056 | 0.373 | 50 |
| 55 | 23.8 | 9.1 | 0.02730 | 3.73 | 36.63 | 0.268 | 30.0 | 190.5 | 160.5 | 0.061 | 0.373 | 55 |
| 60 | 26.3 | 11.6 | 0.02743 | 3.40 | 36.45 | 0.294 | 33.0 | 192.5 | 159.5 | 0.067 | 0.374 | 60 |
| 65 | 28.9 | 14.2 | 0.02759 | 3.12 | 36.24 | 0.321 | 36.0 | 194.5 | 158.5 | 0.072 | 0.374 | 65 |
| 70 | 31.6 | 16.9 | 0.02773 | 2.88 | 36.06 | 0.347 | 38.5 | 196.0 | 157.5 | 0.078 | 0.375 | 70 |
| 75 | 34.5 | 19.8 | 0.02789 | 2.65 | 35.86 | 0.377 | 41.5 | 198.0 | 156.5 | 0.083 | 0.375 | 75 |
| 80 | 37.6 | 22.9 | 0.02805 | 2.46 | 35.65 | 0.407 | 44.5 | 199.5 | 155.0 | 0.089 | 0.376 | 80 |
| 85 | 40.9 | 26.2 | 0.02821 | 2.28 | 35.45 | 0.439 | 47.5 | 201.5 | 154.0 | 0.094 | 0.376 | 85 |
| 86 | 41.6 | 26.9 | 0.02825 | 2.24 | 35.40 | 0.446 | 48.5 | 202.0 | 153.5 | 0.095 | 0.376 | 86 |
| 90 | 44.5 | 29.8 | 0.02838 | 2.10 | 35.24 | 0.476 | 51.0 | 203.0 | 152.0 | 0.100 | 0.377 | 90 |
| 95 | 48.2 | 33.5 | 0.02854 | 1.96 | 35.04 | 0.510 | 54.0 | 205.0 | 151.0 | 0.105 | 0.377 | 95 |
| 100 | 52.2 | 37.5 | 0.02870 | 1.81 | 34.84 | 0.552 | 57.0 | 206.5 | 149.5 | 0.111 | 0.378 | 100 |
| 105 | 56.4 | 41.7 | 0.02889 | 1.70 | 34.62 | 0.588 | 60.5 | 208.5 | 148.0 | 0.117 | 0.380 | 105 |
| 110 | 60.8 | 46.1 | 0.02906 | 1.58 | 34.41 | 0.633 | 63.5 | 210.5 | 147.0 | 0.122 | 0.380 | 110 |
| 115 | 65.6 | 50.9 | 0.02925 | 1.48 | 34.19 | 0.676 | 66.5 | 212.0 | 145.5 | 0.128 | 0.381 | 115 |
| 120 | 70.8 | 56.1 | 0.02945 | 1.38 | 33.96 | 0.725 | 70.0 | 213.5 | 143.5 | 0.134 | 0.382 | 120 |
| 125 | 76.0 | 61.3 | 0.02966 | 1.30 | 33.72 | 0.769 | 73.5 | 215.5 | 142.0 | 0.139 | 0.382 | 125 |
| 130 | 81.4 | 66.7 | 0.02986 | 1.21 | 33.49 | 0.826 | 76.5 | 217.0 | 140.5 | 0.145 | 0.384 | 130 |
| 135 | 87.0 | 72.3 | 0.03009 | 1.14 | 33.23 | 0.877 | 80.0 | 219.0 | 139.0 | 0.151 | 0.385 | 135 |
| 140 | 92.6 | 77.9 | 0.03032 | 1.07 | 32.98 | 0.934 | 83.5 | 221.0 | 137.5 | 0.157 | 0.386 | 140 |

* Inches of mercury below one standard atmosphere (29.92 in.).

## TABLE 62

### Thermodynamic Properties of Isobutane ($C_4H_{10}$)

| Temperature, Degrees F. $t$ | Pressure, Pounds per Square Inch | | Specific Volume, Cubic Feet per Pound | | Density, Pounds per Cubic Foot | | Heat Content, B.t.u. per Pound | | Latent Heat, B.t.u. per Pound $L$ | Entropy, B.t.u. per Pound, Deg. F. | | Temperature, Degrees F. $t$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Absolute $p$ | Gage $g.p.$ | Liquid $v$ | Vapor $V$ | Liquid $1/v$ | Vapor $V$ | Liquid $h$ | Vapor $H$ | | Liquid $s$ | Vapor $S$ | |
| −20 | 7.50 | 14.6* | 0.02610 | 10.5 | 38.35 | 0.0952 | −9.0 | 156.5 | 165.5 | −0.020 | 0.356 | −20 |
| −15 | 8.30 | 13.0* | 0.02620 | 9.90 | 38.15 | 0.101 | −7.0 | 157.0 | 164.0 | −0.015 | 0.354 | −15 |
| −10 | 9.28 | 11.0* | 0.02635 | 8.91 | 37.95 | 0.112 | −4.5 | 168.5 | 163.0 | −0.010 | 0.353 | −10 |
| − 5 | 10.4 | 8.8* | 0.02645 | 7.99 | 37.80 | 0.125 | −2.5 | 159.5 | 162.0 | −0.005 | 0.351 | − 5 |
| 0 | 11.6 | 6.3* | 0.02660 | 7.17 | 37.60 | 0.139 | 0 | 160.5 | 160.5 | 0.000 | 0.350 | 0 |
| + 5 | 13.1 | 3.3* | 0.02675 | 6.41 | 37.40 | 0.156 | +2.5 | 162.0 | 159.5 | 0.005 | 0.348 | + 5 |
| 10 | 14.6 | 0.2* | 0.02690 | 5.75 | 37.20 | 0.174 | 5.0 | 163.5 | 158.5 | 0.011 | 0.348 | 10 |
| 15 | 16.3 | 1.6 | 0.02705 | 5.18 | 37.00 | 0.193 | 7.5 | 164.5 | 157.0 | 0.016 | 0.347 | 15 |
| 20 | 18.2 | 3.5 | 0.02715 | 4.68 | 36.80 | 0.214 | 10.0 | 166.0 | 156.0 | 0.021 | 0.346 | 20 |
| 25 | 20.2 | 5.5 | 0.02730 | 4.24 | 36.60 | 0.236 | 13.0 | 167.5 | 154.5 | 0.027 | 0.346 | 25 |
| 30 | 22.3 | 7.6 | 0.02745 | 3.86 | 36.40 | 0.259 | 15.5 | 169.0 | 153.5 | 0.032 | 0.346 | 30 |
| 35 | 24.6 | 9.9 | 0.02760 | 3.52 | 36.20 | 0.284 | 18.0 | 170.5 | 152.5 | 0.038 | 0.346 | 35 |
| 40 | 26.9 | 12.2 | 0.02780 | 3.22 | 36.00 | 0.311 | 21.0 | 172.0 | 151.0 | 0.044 | 0.346 | 40 |
| 45 | 29.5 | 14.8 | 0.02795 | 2.96 | 35.80 | 0.338 | 24.0 | 174.0 | 150.0 | 0.049 | 0.346 | 45 |
| 50 | 32.5 | 17.8 | 0.02810 | 2.71 | 35.60 | 0.369 | 27.0 | 175.5 | 148.5 | 0.055 | 0.346 | 50 |
| 55 | 35.5 | 20.8 | 0.02825 | 2.49 | 35.40 | 0.402 | 30.0 | 177.5 | 147.5 | 0.061 | 0.347 | 55 |
| 60 | 38.7 | 24.0 | 0.02840 | 2.28 | 35.20 | 0.439 | 33.0 | 179.0 | 146.0 | 0.067 | 0.348 | 60 |
| 65 | 42.2 | 27.5 | 0.02855 | 2.10 | 35.00 | 0.476 | 36.5 | 181.0 | 144.5 | 0.073 | 0.349 | 65 |
| 70 | 45.8 | 31.1 | 0.02875 | 1.94 | 34.80 | 0.515 | 39.5 | 183.0 | 143.5 | 0.079 | 0.350 | 70 |
| 75 | 49.7 | 35.0 | 0.02890 | 1.79 | 34.60 | 0.559 | 43.0 | 185.0 | 142.0 | 0.086 | 0.351 | 75 |
| 80 | 53.9 | 39.2 | 0.02910 | 1.66 | 34.35 | 0.602 | 46.5 | 187.0 | 140.5 | 0.092 | 0.352 | 80 |
| 85 | 58.6 | 43.9 | 0.02930 | 1.54 | 34.10 | 0.649 | 50.0 | 189.0 | 139.0 | 0.098 | 0.353 | 85 |
| 86 | 59.5 | 44.8 | 0.02935 | 1.52 | 34.10 | 0.658 | 50.5 | 189.5 | 139.0 | 0.099 | 0.354 | 86 |
| 90 | 63.3 | 48.6 | 0.02950 | 1.42 | 33.90 | 0.704 | 53.5 | 191.0 | 137.5 | 0.105 | 0.356 | 90 |
| 95 | 68.4 | 53.7 | 0.02965 | 1.32 | 33.70 | 0.758 | 57.5 | 193.5 | 136.0 | 0.112 | 0.358 | 95 |
| 100 | 73.7 | 59.0 | 0.02990 | 1.23 | 33.45 | 0.813 | 61.0 | 195.5 | 134.5 | 0.118 | 0.359 | 100 |
| 105 | 79.3 | 64.6 | 0.03005 | 1.14 | 33.25 | 0.877 | 65.0 | 198.0 | 133.0 | 0.125 | 0.360 | 105 |
| 110 | 85.1 | 70.4 | 0.03030 | 1.07 | 33.00 | 0.935 | 69.0 | 200.0 | 131.0 | 0.132 | 0.362 | 110 |
| 115 | 91.4 | 76.7 | 0.03050 | 0.990 | 32.80 | 1.01 | 73.0 | 202.5 | 129.5 | 0.139 | 0.364 | 115 |
| 120 | 98.0 | 83.3 | 0.03075 | 0.926 | 32.50 | 1.08 | 77.0 | 204.5 | 127.5 | 0.147 | 0.367 | 120 |
| 125 | 104.8 | 90.1 | 0.03095 | 0.867 | 32.30 | 1.15 | 81.5 | 207.5 | 126.0 | 0.154 | 0.369 | 125 |
| 130 | 112.0 | 97.3 | 0.03125 | 0.811 | 32.00 | 1.23 | 86.0 | 210.0 | 124.0 | 0.161 | 0.371 | 130 |
| 135 | 119.3 | 104.6 | 0.03145 | 0.760 | 31.80 | 1.32 | 90.5 | 212.5 | 122.0 | 0.169 | 0.375 | 135 |
| 140 | 126.8 | 112.1 | 0.03175 | 0.716 | 31.80 | 1.32 | 95.0 | 215.5 | 120.5 | 0.176 | 0.377 | 140 |

* Inches of mercury below one standard atmosphere (29.92 in.).

TABLE 63

ON THE BASIS OF 200 B.T.U. REFRIGERATION PER MINUTE AND 86 DEGREES CONDENSING AND 5 DEGREES EVAPORATING TEMPERATURE

| $C_2H_2Cl_2$ | Weight of Refrigeration, Pounds | Volume in Cubic Feet per Minute | In Inch of Mercury | | Ratio $\frac{p_2}{p_1}$ | Coefficient of Performance | Horse Power per Ton |
|---|---|---|---|---|---|---|---|
| | | | $p_1$ | $p_2$ | | | |
| | 1.768 | 108.4 | 1.78 | 14.65 | 8.23 | 5.14 | 0.918 |

TABLE 64

PROPERTIES OF SATURATED ETHANE

| Temperature, Degrees F. | Pressure, Pounds per Square Inch | | Heat Content above 32 Deg. F., B.t.u. per Pound | | Latent Heat, B.t.u. per Pound, Total |
|---|---|---|---|---|---|
| | Gage | Absolute | Liquid | Vapor | |
| −40.0 | 100.0 | 114.7 | −32.4 | 149.6 | 182.0 |
| −30.0 | 120.3 | 135.0 | −27.9 | 150.4 | 178.4 |
| −20.0 | 144.8 | 159.5 | −23.4 | 150.6 | 174.0 |
| −10.0 | 173.3 | 188.0 | −18.9 | 150.3 | 169.2 |
| 0.0 | 204.3 | 219.0 | −14.4 | 149.4 | 163.8 |
| 10.0 | 238.3 | 253.0 | − 9.9 | 148.1 | 158.0 |
| 20.0 | 277.3 | 292.0 | − 5.4 | 145.7 | 151.1 |
| 30.0 | 318.3 | 333.0 | − 0.9 | 142.1 | 143.0 |
| 40.0 | 361.3 | 376.0 | 4.0 | 137.1 | 133.1 |
| 50.0 | 410.3 | 425.0 | 9.4 | 130.8 | 121.4 |
| 60.0 | 466.3 | 481.0 | 15.2 | 122 1 | 106.9 |
| 70.0 | 532.3 | 547.0 | 22.2 | 111.4 | 89.2 |
| 80.0 | 610.3 | 625.0 | 32.3 | 98.2 | 65.9 |
| 89.8 | 703.4 | 718.1 | ..... | ..... | 0.0 |

**Mixtures of Different Refrigerants.**—Various attempts have been made to mix different refrigerants with the idea that a new " compound " will be obtained or something that will give the characteristics in between the extremes of the two being mixed. This has been tried with carbon dioxide and sulphur dioxide, methyl and ethyl chloride, and butane and propane. Such attempts have always been failures, as a new compound is *never* obtained, and leaks occurring soon change the proportions of the mixture. In fact, if leaks do occur, it is the vapor with the higher tension that escapes in the largest proportion. As the mixture is not a pure

compound, and the proportion is not known accurately, pressures in the system cannot be interpreted with any accuracy.

**Water Vapor.**—Water vapor has been proposed for years, especially for air cooling where the refrigerating temperature needs to be between 45 and 60 deg. F. If steam is used as a refrigerant under these conditions, the evaporating pressures have to be extremely low, and the specific volume of the saturated vapor is very great. The only method seriously proposed for this condition is the Leblanc system of compressing with water pistons, similar to the method of securing high vacuum in steam condensers. Although a number of applications to warship and other refrigeration has been made in France, little has been attempted in the United States. The advantage would be perfect safety, the "brine" used being pure water, yet the horse power per ton is considerably higher than would be the case with other refrigerants.

**Air.**—Air as a refrigerant was one of the first successful means of securing mechanical refrigeration, and it has many good features. Air is cheap and plentiful and leaks are harmless except from the point of view of the loss of capacity or the cost in dehumidifying it. Yet it has been discontinued in all except a very few installations because of the bulk of the compressor, operating troubles, and the higher horse power per ton of refrigeration than would be necessary with the volatile liquids now used so generally. In order to secure low temperatures an expansion cylinder *must* be used which returns some of the power required for compression back to the shaft. In order to decrease the piston displacement per ton of refrigeration a *dense* air usually is used, of three or more atmospheres' suction pressure, in which case the cycle is a closed one. Although there are decided advantages in the use of air, yet it seems unlikely that one would face the operating troubles with the lubrication at the very low temperatures, from −50 to −100 deg. F., and with the continued trouble with the trace of water vapor which cannot be removed from the air and which will freeze in the valves of the expansion cylinder at any moment; a condition which occurred almost daily in the dense air machines in the U. S. navy. Its use in the future is very doubtful when it is considered that very much better results can be obtained by the use of the many available refrigerants of the volatile liquid type (see Chapter I).

**Lubrication.**—Lubrication of the compressor cylinder is one of the very important points to be considered. The difficulty in refrigeration is that the refrigerant frequently is absorbed by the oil or vice versa, or that some chemical action takes place. The lubricant must not freeze at the lowest temperature used, usually taken at −20 deg. F., and must not show a flash test at less than 300 deg. F. Experience has shown

that a straight, refined, mineral oil with a viscosity of about 130 to 180 Saybolt at 104 deg. F. is about correct for most work. The presence of water[8] in the ammonia or the oil will tend towards an emulsifying effect due to the ammonia hydrate formed.

In the case of carbon dioxide, glycerine was used for some time, but now a mineral oil is used exclusively in the practice in the United States and Great Britain. On account of the low temperatures so often encountered with carbonic compressors, a much lower test oil is required than for ammonia. Chemically pure, dry glycerine has been considered to be the only practical lubricant for ethyl and methyl chloride, whereas sulphur dioxide requires a straight refined, highly anhydrous, mineral oil. The hydro-carbon liquids tend to dissolve mineral oils, and where the liquid gets into contact with the oil to any extent it is found that glycerine or one of the glycols mixed with deflocculated graphite will give good results.

**Tables and Charts.**—Figures 164 to 168, inclusive, are P–I diagrams for carbon dioxide, ammonia, ethyl chloride and methyl chloride, respectively. These diagrams (see Chapter I) represent the action of the entire refrigerating cycle and they present an easy and accurate method for the solution of problems in refrigeration. Tables 59, 61, 62 and 64 were compiled by the Linde Air Products Company.[9] Table 56 and Fig. 166 are from D. L. Fiske, Refrigerating Engineering, Dec., 1924. The properties of saturated and superheated ammonia, and the P–I diagram for ammonia are taken from Circular 142 of the Bureau of Standards.

[8] W. F. Osborne, Journal of the Amer. Soc. Refrig. Eng., 1920.

[9] Refrigerating Engineering, June, 1926.

## CHAPTER VIII

## BRINE AND BRINE SYSTEMS

As before noted, refrigeration may take place *directly* through the absorption of heat by the refrigerant from the commodity to be cooled, and *indirectly* where water or some non-freezing solution is cooled by the refrigerant and then in turn cools the commodity by means of pipe coils or by sprays. The small job generally uses the direct method (called *direct* expansion), while for larger work whether the direct or the indirect (the brine system) shall be used depends on a number of factors, so that each case has to be decided separately on its merits.

**Costs.**—With direct expansion piping for ammonia it is necessary to have an extremely tight system. With steam and compressed air piping a small leak is usually of little moment, but in the case of ammonia even a small leak may be very serious, both as regards the cost of the refrigerant lost and also because of the possible danger to persons and commodities from the released ammonia. To keep the pipe line tight means special fittings and very careful pipe work, with either the tinned and soldered or the lithage and glycerine thread joint.

On the other hand, the brine refrigerating piping need not be special. The pressures used are generally not greater than 25 lb. in excess of the static pressure due to the head of brine, and the piping and the fittings can be the usual kind for 125 lb. pressure. If the brine system is a closed one with a balance tank (Fig. 169), the pumping cost need be only that which is required to overcome friction and create the velocity head. There is required, however, some kind of brine cooler of the double pipe, the shell and coil, but usually of the shell and tube variety, and the necessary circulating pump for the brine. As the brine is used to *absorb* heat and its heat capacity is limited to a rise in temperature of a few degrees only, it follows that the brine circulating pump must be in operation as long as refrigeration is required, unless hold-over or congealing tanks or brine tubes are used to hold a quantity of brine and thereby store up a relatively small amount of refrigeration.

Direct expansion means, in plants of any size, a heavy initial cost for the charge of ammonia or other refrigerant. Should a large leak occur, due to a broken fitting or pipe, there is danger of the entire charge being

lost. With brine it is also possible to lose the charge, but if the piping is open to inspection there should not be much difficulty in keeping the loss small.[1]

The refrigerating machine can be operated periodically when brine and a brine storage tank are used, but the brine pump needs to operate continuously. In direct expansion the compressor has to pump as long as refrigeration is desired.

**Advantages of Brine.**—There is an advantage in the use of brine from the fact that it may be used as a storage for refrigeration and therefore the brine-operated plant with brine storage can respond to sudden demand, as, for example, in the quenching of steel, in the pasteurizing of milk or in other cases where the cooling required comes in

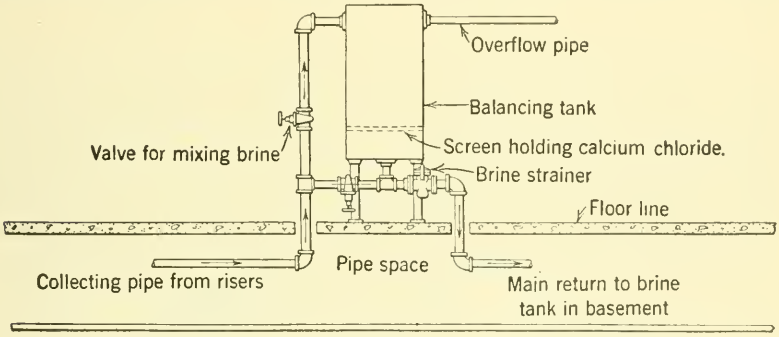



Fig. 169.—A Typical Connection to a Brine Balancing Tank.

sudden demands. Under these conditions it is possible to use a brine storage tank of size sufficient to obtain the required storage capacity, in which case a smaller refrigerating machine may be operated practically continuously. When the temperatures required are not below the temperature of freezing of water (32 deg. F.), it is usual to substitute water for the brine, unless (as in the case of steel quenching) there are good reasons for the use of calcium or sodium chloride brine. Some kinds of work are best performed by employing a spray (particularly in the case of air cooling and conditioning), using a specially designed spray nozzle so as to atomize the brine or water as finely as possible. The special advantage here is in the washing of the air, and—for the lower temperatures—in the condensation of the moisture in the air so as to make the moisture content correspond to the dew point temperature without the freezing of the moisture on the pipes. It has been found also that a quicker and more satisfactory cooling can be obtained

[1] See District Cooling, Chapter XIV.

at times by the use of sprays[2] due to the intimate contact of the spray with the air. For example, in the packing house where the hot carcasses must be cooled quickly, uniformly, and with least shrinkage in the weight, it has been found that the brine spray is the best to be used. The spray deck is designed so that the entire air volume of the room in which the hot meats are hung can be changed (or recirculated) once a minute or once per one-half minute. The sprays drag the air in the direction of their flow and the nozzles are so placed as to provide a curtain of liquid through which the air has to pass. The air is thereby cooled to the temperature of the brine or of the water. (See Chapter XIV.) Water sprays are made use of extensively in the cooling of theatres and auditoriums, in chocolate and gelatine factories, or wherever a temperature of 35 degrees or more can be employed with satisfaction.

There are some industries which cannot permit the use of ammonia in the process plant. Carbon dioxide usually is innocuous, but carbon dioxide is not in frequent use in the United States. In England and in Northern Europe the condensing water is usually quite cool and the result is that carbon dioxide can be used with satisfaction, and is used for about 50 per cent of the time, whereas in India and other tropical countries with water temperatures in the neighborhood of 100 deg. F. or higher the refrigerant may be sulphur dioxide or ethyl chloride. Where leaks of ammonia may cause considerable damage, and careless labor makes the danger of accidents a live one (as is the case in some industrial plants) it may be best to use the indirect system, employing of brine or refrigerated water. However, it must be kept in mind that the indirect brine system is always operated at greater expense, as in cooling brine for use in refrigerating piping the boiling temperature of the ammonia must be, usually, 10 deg. F. lower than the boiling temperature of the ammonia when the direct expansion piping is used. This is because of the double cooling that is required, although when sprays are used it is doubtful whether this statement is strictly correct. The brine has to be circulated (which also assists in heating the brine by an amount equal to the heat equivalent of the power used), and at times this power item is an important one. In direct expansion the liquid ammonia will pass to the expansion valve because of its own static pressure due to the condenser pressure, and the suction line, which usually is relatively short, is designed so as to reduce the pipe friction loss to a minimum. Usually it is necessary to restrict the length of

[2] With brine sprays, especially if the temperature is low, foaming may result, in which case the surface tension of the brine needs to be increased. This can be done by the addition of a little kerosene plus oleaic acid.

the suction return line, and so when the compressor has to be remote from the location of the process requiring cooling, the brine system has to be used.

**Brine Cooling.**—In cooling brine there is a decided advantage in using the shell and tube brine cooler (Figs. 170 and 171). When well

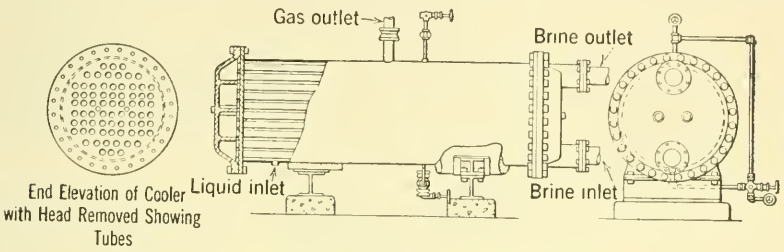



Fig. 170.—Shell and Tube Brine Cooler for Eight Passes.

designed and constructed it is simple, compact and durable, and large capacity is obtained with efficiency of surface and with little trouble from leaks. As a rule a surface of 14 to 15 sq. ft. per ton of refrigeration is used. The construction is very similar to that of the steam surface

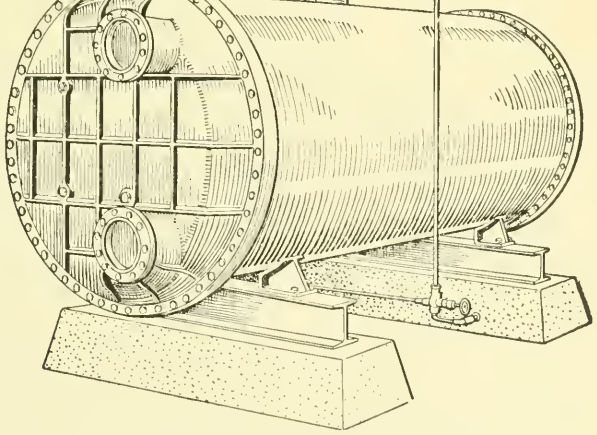


Fig. 171.—Shell and Tube Brine Cooler.

condenser, and the horizontal return tubular boiler. The diameter is usually limited to from 48 to 54 in., and the shell thickness varies from $\frac{1}{4}$ to $\frac{7}{16}$ of an inch, depending on the diameter of the shell, while the tube sheet is usually $\frac{1}{2}$ in. The tubes are usually 2-in. normal diameter.

and are of No. 12 gage charcoal iron, and are expanded into the tube sheet. The design may permit a single pass of the brine, as is the case of the ice-making installations, where the cooler is submerged in the brine tank, or there may be eight or more passes by the use of baffles in the heads. As a rule the number of passes should be kept down to six, for one of the advantages of the use of the brine cooler is in the high value of the coefficient of heat transfer which is due in part to the rapid flow of the brine through the tubes. If the number of passes becomes too great, the resistance to the brine flow may become excessive and costly. Examples are known where the pressure difference between the entrance and the exit has been as much as 75 lb. per sq. in.

Shell and tube coolers are liked because the ammonia part may be made very self-contained. With the possible exception of the condensers (which may be on the roof), the whole system may be placed in the

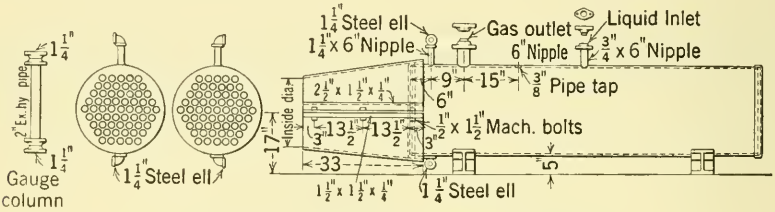



Fig. 172.—Single Pass Brine Cooler.

compressor room under the eye of the engineer. The ammonia charge is not so great and the pressure drop from the evaporator surfaces can be made almost inappreciable. Only one expansion valve need be used (per cooler), and the liquid is best carried at about three quarters full. A liquid trap can be placed in the suction line about six feet above the top of the cooler to separate the liquid ammonia entrained in the suction gas, and this liquid may be returned to the cooler through a pipe line, using a goose-neck connection.

The advantages of the horizonal over the vertical shell and tube cooler are as follows: The ammonia in the horizontal cooler does not have the free passage out as does the vertical one, and the liquid priming is caught by the tubes which are staggered to some extent. The head-room required is not so great, although more space is required on the floor, but there is greater ease in any repair work that is necessary. These coolers should have connections at the top and the bottom for the gage glass, an outlet at the bottom for a blow-off and the liquid ammonia inlet, and at the top for the gas outlet connection. The shell and the tube brine cooler has the disadvantage of danger due to the freezing of the

brine, and the consequent possible bursting of the tubes. It is advised that the calcium chloride brine only be used, because of the much lower freezing temperature of this brine, and that a careful check be kept of the density. Even under these conditions the stoppage of the brine pump may cause trouble, if the ammonia boiling temperature is such as will be below the freezing temperature of the brine.

The design of the brine pipe system (Figs. 174, 175, 176, and 177) is not very well standardized. The essential feature is to reduce the pumping cost to a minimum. This means, wherever possible, the use of an expansion tank at the high point and of a piping arrangement so

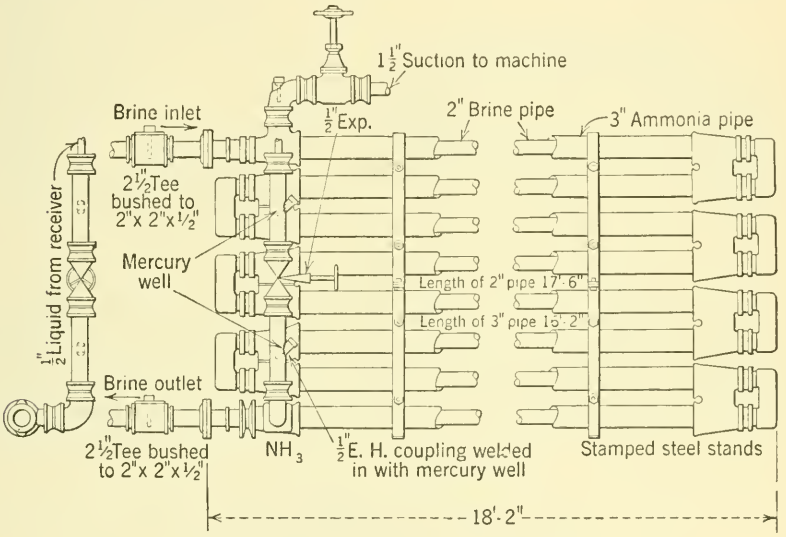

1½" Suction to machine
Brine inlet
½" Exp.
2" Brine pipe
3" Ammonia pipe
2½" Tee bushed to 2" x 2" x ½"
Mercury well
1" Liquid from receiver
Length of 2" pipe 17'-6"
Length of 3" pipe 15'-2"
Brine outlet
2½" Tee bushed to 2" x 2" x ½"
NH3
½" E. H. coupling welded in with mercury well
Stamped steel stands
18'-2"

Fig. 173.—Double Pipe Brine Cooler.

that the brine operation in a complete circuit will be of the nature of a syphon. Under these conditions the pump work consists only of the friction and velocity head. The static pressure on the pump, of course, is due to that required to overcome friction plus the head of brine from the pump to the expansion tank.

**Corrosion of Brine.**—Considerable attention has been directed towards the causes, and the means of their reduction, of corrosion in the brine system. Corrosion may be the result of a number of causes due to the fact that all brines are electrolites, which under certain conditions may stimulate galvanic action. This galvanic action may be caused by dissimilar metals or even by the difference in the potential in different parts of the same metal due to stresses developed in the manufacture or the assembling of the material. Mill scale will sometimes

start corrosion as will also "stray" direct currents on their return to the generator in the power plant, or the presence of air in solution in the liquid may be the cause of the trouble.

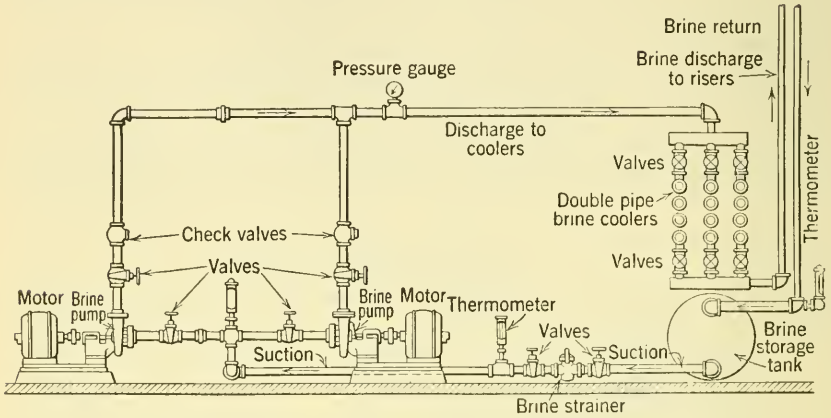



Fig. 174.—Brine Cooler, Brine Tank and Pump.

From the preceding it is evident that if corrosion is to be reduced to a minimum these factors must be removed or decreased in relative importance, and such brine should be used as will give the best results for the work being carried on. At present there are three brines in general

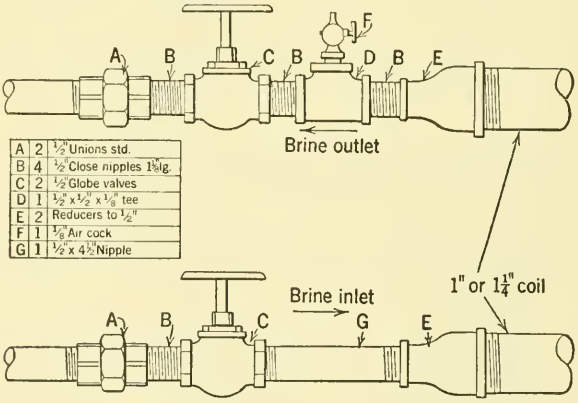



Fig. 175.—Typical Connections to Brine Coils.

use: the sodium or salt brine, the calcium chloride and the mixed magnesium-calcium chloride brines. Which of these different kinds of brines shall be used depends on the kind of service to be rendered. For

example, the packers using spray nozzles to a large extent in the chill rooms for chilling the warm carcasses employ salt brine practically entirely for that service, because of the greater "drying" action of calcium, its greater cost, etc. Calcium chloride is used particularly where low temperatures are desired, or where it is thought that there is danger of the freezing of the brine cooler, in which case there is greater safety in using a brine of lower freezing temperature than is possible with salt brine.

According to tests by Morgan B. Smith,[3] all brines should be tested

[3] Morgan B. Smith, A. S. R. E., 1911:

"The protection of refrigerating apparatus against corrosion involves three main heads, the metallic apparatus, the brine solution and the relation of the refrigerating apparatus proper to the other apparatus in the vicinity, especially with regard to the generation of electrical current. With regard to the refrigerating apparatus, several factors must be considered, i.e., first, the nature of the metals used with consideration of their ability alone to resist corrosion, and their tendency in combination to set up galvanic action and induce corrosion; second, the design of the apparatus and the general workmanship in assembling the varied elements which go to form the finished plant.

"The nature of the metals used and their combination is worthy of very close study as both practice and experimentation have proved. For example, if we place two electrically dissimilar metals, such as brass and iron, in a brine solution an electric current is at once set up—such a current passing from the iron to the brass through the brine solution. The action of this current is such that the iron gradually passes into solution and subsequently is oxidized to form the familiar oxide called rust. The solution of the iron in such a case may take years. On the other hand, it may take a much shorter time, for the rapidity with which such a reaction takes place varies directly with the magnitude of the current set up. This is equally true of the combination of any electrically dissimilar metals, only varying with the relative degree of dissimilarity in the metals used, and for this reason it is perfectly obvious that but one metal should be used as far as possible in refrigerating plants.

" Not only should but one metal be used as far as possible but that metal should be of the greatest possible purity, for impurities in metals act in much the same manner as do two dissimilar metals mechanically combined. This is especially true of cast iron, which is frequently destroyed by the numerous local currents set up between the iron and its impurities leaving the casting in a characteristic graphitic condition. It is well at this time to consider a widely used combination of two electrically dissimilar metals, namely, galvanized iron.

" Zinc resists corrosion in a brine solution if no electrically dissimilar metal be present. If, however, iron is present the zinc will go into solution, thereby protecting the iron as long as zinc is present in sufficient amounts. Therefore it is plain that galvanized iron should never be used excepting when of the very best quality, in which no iron is present that is exposed to the brine. When galvanized iron does show iron exposed at pin holes, or along seams where bending has cracked the zinc film it is next to worthless in brine solutions, for sooner or later all the zinc will be stripped off leaving the iron bare. As there is evidence that the highest grade of iron is not used in the manufacture of galvanized iron, it is plain that the iron itself will be more liable to corrosion than would a plain iron apparatus of a high grade of purity. It is doubtful, therefore, if galvanized iron ever should be used since it seems almost impossible to construct apparatus of this material so that only zinc will be exposed to the brine solution.

" Many alloys resist corrosion to a marked degree, especially the bronzes. Alloyed copper in iron tends to retard corrosion, whereas unalloyed copper in iron greatly hastens corrosion. (Eng. News, Aug. 3, 1911.) From the foregoing facts it is very evident that as far as practical one homogeneous metal only should be used in the construction of apparatus.

" The design of apparatus, especially of fittings and piping, is very important and is generally wholly neglected. When a piece of metal is strained an electrical difference of potential is at once set up between the strained and the unstrained portions, and it is the strained portion which is attacked and eventually destroyed. This state of affairs is exemplified in the case of threaded pipe ends, engaging fittings, the pipe ends being strained while the remainder of the pipe is unstrained. There can be no doubt whatever that this state of strain in pipe ends is largely responsible for the failure of piping at the threaded portions. This is almost equally true of fittings which,

from time to time for acidity, as acidity increases very greatly the process of corrosion. In general he advises the use of some alkali like lime (CaO) or caustic soda, or the use of a bag of lime suspended in the tank or at some other convenient location. In his opinion, "the least corrosive chloride is that *calcium chloride* which contains as small an amount of magnesium and sodium chloride and the least practical amount of water." He also says that there is no longer any doubt that magnesium and sodium chloride increase corrosion when present in

however, generally have so much more metal remaining after the threads are cut that the strain caused in them when making up joints cannot be so great as it is in piping where comparatively little metal is left after threading. There are a number of instances where pipe trouble has been eliminated by the use of piping and fittings of extra weight, so that after the threads are cut there is left an abundance of metal sufficient to withstand the making up of joints without undue strain.

"Air is one of the great accelerators of corrosion, and it is very essential to see that no air pockets can form in the apparatus, especially in the piping. All high points in the piping and each piece of apparatus as far as practicable should be provided with suitable vents so that any trapped air can be withdrawn from time to time. If an air pocket does exist for some time, corrosion soon makes itself evident, and if allowed to go on will eventually cause failure at the point.

"We may now consider the brine solution to be used. Since the brine is made up by dissolving certain chlorides in water, it is very evident that the nature of the water is sure to affect the resulting brine solution. Water containing acid of any description should never be used, since the resulting brine will also be acid. An acid brine is far more corrosive than an alkaline or a neutral brine.

"As a matter of fact all brine solutions should be tested from time to time for acidity, and if found to be acid should be neutralized at once with some alkali such as lime or caustic soda. Sufficient alkali should be added to make the brine distinctly alkaline towards a suitable indicator. The best indicator for such purpose is *phenol-phthalein*. This is made up by dissolving phenol-phthalein in alcohol—about one-half ounce in two quarts—and adding to the solution two quarts of distilled water. Another good indicator which may be used is methyl-orange, which turns yellow on adding brine if the brine is alkaline and red if the brine is acid. When making up the brine it is well to avoid saturating the brine with air, for brine saturated with air is far more corrosive than brine carrying only a minimum amount of air in solution. Brine should never be agitated with an air-blast, nor should it be allowed to fall through the air into tanks as is often the case with return brine. Return brine is often very nearly saturated with air in just this manner. As for the strength of the brine and the temperature it is certain that the greater the concentration and the lower the temperature the less likely is the brine to assist corrosion.

"The selection of chloride to use in making up the brine solution is no longer a serious problem since practice and experimentation have proved the truth of the assertion that the most economical and the least corrosive chloride is that calcium chloride which contains as small an amount of magnesium chloride and of sodium chloride as possible, and the least practicable amount of water.

"There is no longer any doubt that magnesium chloride and sodium chloride in calcium chloride brine materially increase the corrosiveness of the resulting brine, compared with a brine containing practically no chloride other than calcium chloride. This is especially true if the brine containing magnesium chloride becomes acid through the unstable nature of magnesium chloride.

" I have been asked whether ammonia present in brine would increase the corrosiveness of the brine. Experience of long duration, in which the ammonia content of the calcium chloride brine in question was maintained at a constant value, shows that *ammonia tends to retard rather than to hasten it*, when present in sufficient amount.

" In addition to the effect of galvanic action caused by different materials in the system corrosion may be greatly hastened by the action of stray electric currents which find a way into the refrigerating apparatus from which they pass to some neighboring metallic circuit or directly to the ground. If the refrigerating apparatus, in whole or in part, is positive in electrical potential towards the ground we may be sure that some stray electric current is finding its way into the apparatus, and that corrosion is being hastened in proportion to the magnitude of the stray current. It is said that a difference of potential of 1/1000 volts is all that is required to start corrosion two miles from the dynamo. It is a wise precaution, therefore, to see that direct current apparatus of all sorts is properly insulated so that it will not be a continual menace to the refrigerating engineer and his plant."

calcium brine. This, he says, is on account of the unstable nature of magnesium chloride, which tends to become acid and cannot be made alkaline.

Emerson P. Poste[4] says, as a result of his own experiments, "essentially pure calcium chloride is alkaline when first made but it turns acid on exposure to air. With this acidity comes a marked increase in the corrosive action. The acidity is materially increased by the presence of magnesium chloride, as the result of the earlier development of acidity and the formation of corrosive ammonium chloride in the case of ammonia leaks. Contact of unlike metals and the presence of these impurities accelerate these electric tendencies. The corrosive action of brine decreases with increasing brine density. Chlorides are on the market which are contaminated with magnesium chloride, though first-class materials are available both as soda by-product and natural chlorides. Corrosive brines free from magnesium chlorides may be corrected by treatment with lime if the alkalinity produced is maintained above 0.1 per cent.

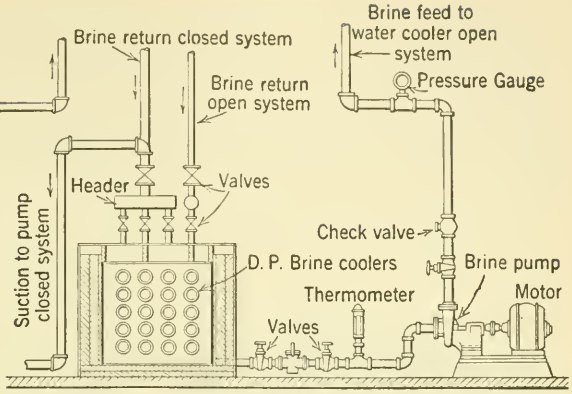



FIG. 176.—Typical Connections to Brine Pump.

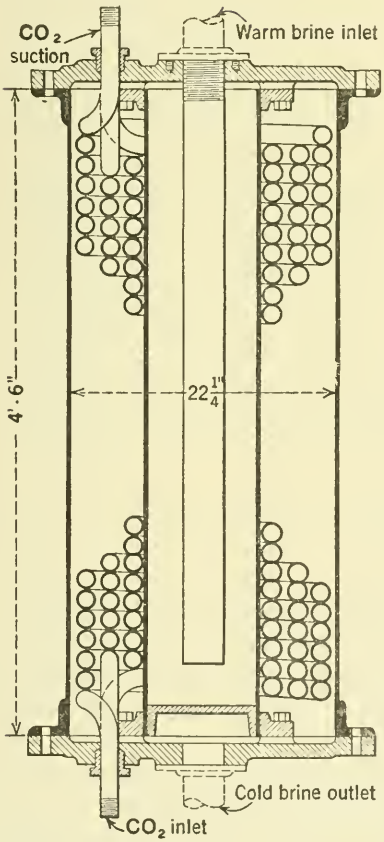



FIG. 177.—Shell and Coil Brine Cooler for Carbon Dioxide.

"The careful plant manager who

[4] Emerson P. Poste, Milk Dealer, January, 1923.

is interested in avoiding operating irregularities, in keeping his equipment in the best of condition and in getting the most out of it will use only the best of chloride in making his brine, will keep the gravity approaching 1.2, will hang a bag of lime in his brine tank and maintain therein a good supply of material, and will avoid unnecessary contact between the brine and the air." Fig. 178, taken from these experiments, shows the relative rates of corrosion.

The presence of ammonia in the brine caused by ammonia leaks frequently causes considerable concern other than because of the loss of the ammonia. Morgan B. Smith claims that experiments of long duration, during which time the ammonia content was maintained constant, show that ammonia tends (as has been said before) to retard rather than to hasten corrosion. Of course, in the case of ammonia pipe leaks, the thing to do is to stop the leaks. Much ammonia in the brine will probably remove the galvanizing from the cans (if the brine is in an ice tank), but after this is removed the corrosion will stop unless ammonium chloride is formed as it will if magnesium chloride is present. Sal ammoniac will accelerate corrosion of iron and steel. If ammonium chloride is not formed it would not seem that any great harm is caused by the presence of an alkaline brine occasioned by the presence of ammonia.

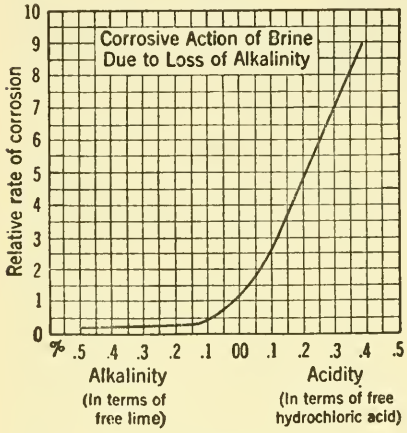



Fig. 178.—Relative Corrosion of Brine

Too much alkalinity is undesirable where galvanized coatings form a large part of the metal exposed to its action,[5] as would be the case in can ice freezing tanks. Experiments prove that commercial calcium chlorides now on the market show decided corrosive action due to their high soluble alkalinity, but weak brines show more action than do the stronger brines of the same degree of alkalinity. Weak brines with ammonia or ammonium chloride present tend to increase the amounts of corrosion.[6]

[5] Arthur C. White, Industrial and Engineering Chemistry, May, 1925.

[6] As the ice can is always galvanized, galvanic action is always present if the brine is acid. In this case it has been shown by A. C. White that "the action of calcium chloride brine on galvanized coatings is such that the galvanizing can be saved by having the least alkalinity possible, and that neutral brines have the least effect."

Should, however, it be decided to remove the ammonia from the brine it is most probable that heating the brine will be found to be the only practical method. In the larger plants, where care is taken in details, and the brine is kept alkaline and not permitted to get weak or to absorb air, corrosion should not be great with any kind of brine, and at times the cheap brine (if much brine has to be wasted) with some corrosion will be found to be more economical than the more expensive kind that is not so corrosive.

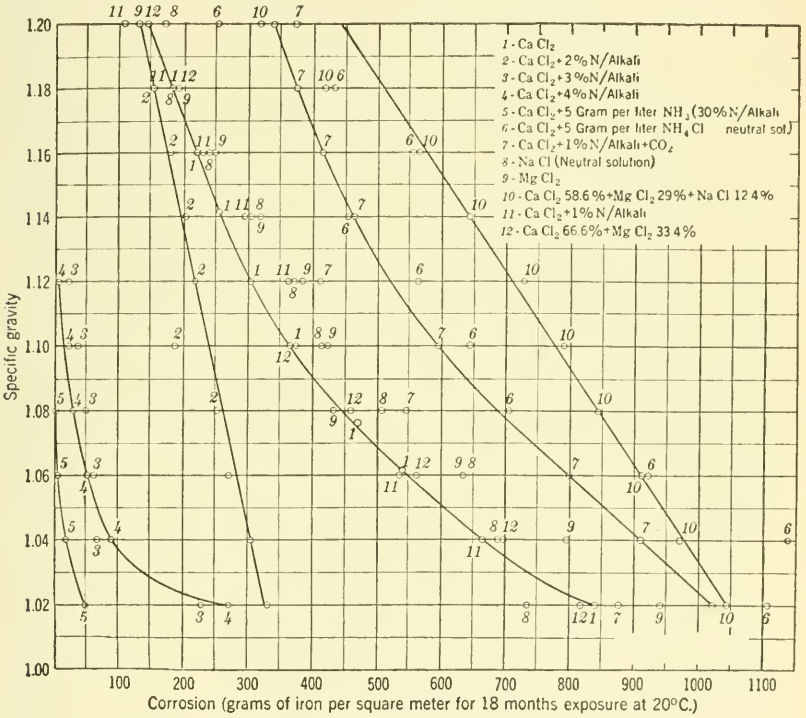



Fig. 179.—Tests on Corrosion of Calcium Chloride Brine.

Figure 179 shows the results of tests by C. A. Cole in the laboratory of the Senet-Solway Company. Twelve series of experiments were run at the same time, each series being made up of 10 tests under the same conditions. In each test a 3-in. piece of $\frac{1}{4}$-in. polished iron taken from the same rod was completely submerged in 100 c.c. of solution. The iron had an analysis of 1.17 per cent C, Mn and Si nearly $\frac{1}{10}$ per cent, and S 0.06 per cent. These results showed that corrosion decreased with the concentration. The solution should be alkaline and should not be exposed to air or to $CO_2$ in the air. As magnesium chloride solution

cannot be made alkaline, such solutions are objectionable because of the increased corrosion. Finally, brine solutions should not contain any considerable amounts of three constituent salts.

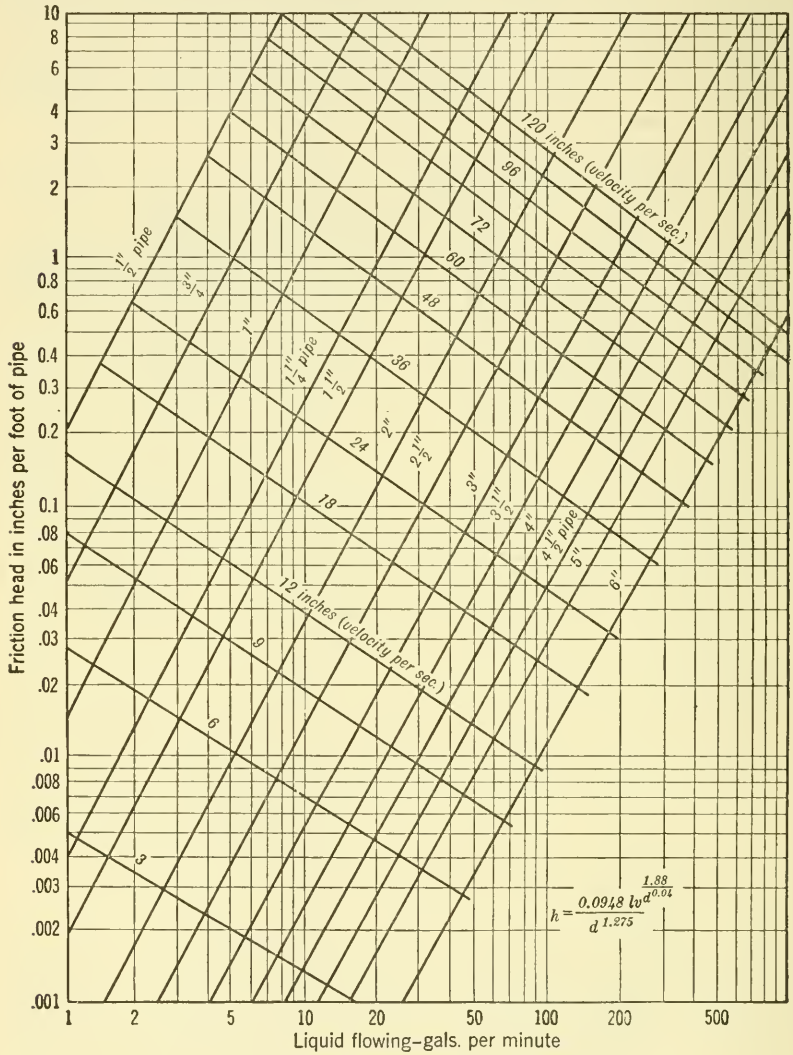



FIG. 180.—Resistance to Flow of Water in Pipes.

**Report of the Corrosion Committee.**—The report of the Corrosion Committee[6a] is abstracted as follows:

[6a] Report of the Corrosion Committee, Refrigerating Engineering, Jan., 1927, by Russell, Roberts and Chappell.

**For Closed Brine Systems.**— Use may be made of sodium dichromate ($Na_2Cr_2O_7+2H_2O$) and sufficient sodium hydrate to convert the dichromate to chromate. For calcium chloride brine use 100 lb. of dichromate per 1000 cu. ft. of brine and add 35 lb. of 76 per cent caustic. For sodium brine use 200 lb. of dichromate per 1000 cu. ft. of brine and add 70 lb. of 76 per cent caustic. The brine should be slightly alkaline after adding the salt and the caustic.

**Open Brine Systems.**— For sodium chloride brines use di-sodium phosphate ($Na_2HPO_4+12H_2O$) at the rate of 100 lb. per 1000 cu. ft. of brine. The brine must not be too alkaline. Test for alkalinity with phenolphthalein and if it shows *pink* hydrochloric acid is to be added. Test strips of bright iron or steel should be suspended in the brine in. order to note the rate of corrosion.

**Zinc Dust.**—Zinc dust may be used to advantage in open calcium chloride systems at the rate of 60 lb. of zinc dust per 1000 cu. ft. of the brine. A little should be added at a time.

**Condensers.**—For fresh water recirculating condenser systems use 17 lb. of 40 degree silicate per day per 1000 cu. ft. of make up water. The water should test pink with phenolphthalein one hour after adding the silicate.

**Friction to Flow of Brine.**—The laws underlying the resistance to flow of brines have been given very little attention. The only published information on the subject is a paper by Professor A. H. Gibson[7] of the

[7] A. H. Gibson, The Resistance to the Flow of Brine Solutions through Pipes, The Institute of Mechanical Engineering, Feb., 1914.

Lord Raleigh has shown that the resistance to flow of a non-compressible fluid can be given by:

$$R \propto \rho v^2 \phi\left(\frac{\mu}{\rho v d}\right) \text{lb. per sq. ft. of surface,}$$

where

$\rho$ = the density of the fluid in pounds per cubic foot;
$\mu$ = coefficient of viscosity of the fluid in foot-pound units;
$v$ = mean velocity of flow;
$d$ = diameter of the pipe;
$\phi\left(\frac{\mu}{\rho v d}\right)$ must be found by experiment.

Osborne Reynolds found by experiment for water flowing in pipes

$$R \propto \rho v^2 \left(\frac{\mu}{\rho v d}\right)^{2-n} \text{lb. per sq. ft. of surface.}$$

*Footnote continued on page 266.*

University College, Dundee. Dr. Gibson states that, as pointed out by Reynolds, with a rough pipe the frictional head loss is independent of

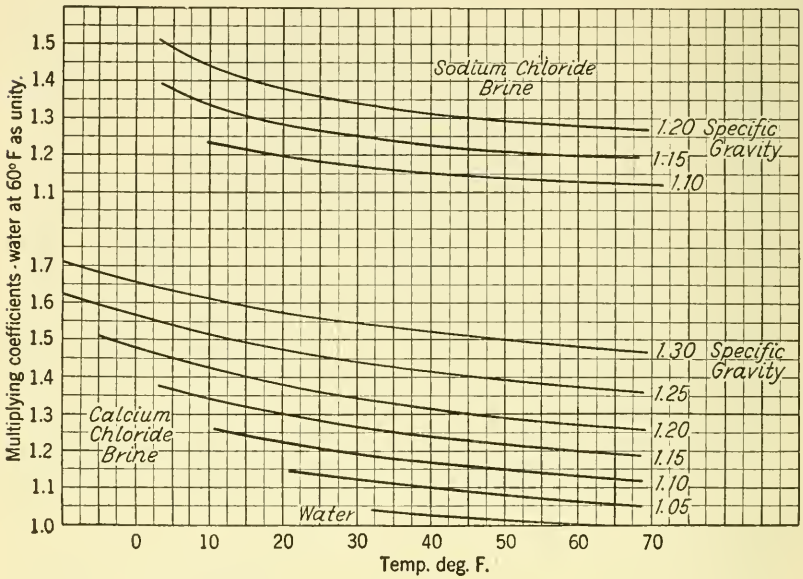



Fig. 181.—Gibson's Multiplying Factors for Brine.

viscosity and that for velocities above the critical the resistance to flow is proportional directly to the densities. Using a Reynolds' number of

Putting $h$ = the loss of head in a pipe of length $l$, expressed in terms of feet of a column of the fluid flowing then as

$$F \propto R \times \text{the wetted surface},$$

$$h\rho\frac{\pi d^2}{4} \propto \rho v^2 \left(\frac{\mu}{\rho v d}\right)^{2-n} \pi d l,$$

or,

$$h \propto \left(\frac{\mu}{\rho}\right)^{2-n} \left(\frac{v^n}{d^{3-n}}\right) l \text{ ft.}$$

Therefore different fluids with identical values for the index $n$ will be related according to

$$\frac{h_1}{h_2} = \left(\frac{w_1}{w_2}\right)^{2-n} \left(\frac{v_1}{v_2}\right)^n, \quad \text{where} \quad w = \frac{\mu}{\rho},$$

and at identical velocities,

$$\frac{h_1}{h_2} = \left(\frac{w_1}{w_2}\right)^{2-n}.$$

1.9 for all pipes, Dr. Gibson calculated multiplying factors for salt and calcium brine. These factors are to be the multipliers of the friction head of water at 59 deg. F. The head lost by water is shown in Fig. 180, and the multiplying factor is given in Fig. 181. However, the results given in these curves appear to be large by an amount varying from 10 to 20 per cent.

**Concrete Brine Tanks.**—The concrete brine tank can be used to advantage at times. According to Boyden,[8] a mixture of 1 : 2 : 4 under 30 ft. of water is waterproof. The concrete must be made of fine clear sand and hard, durable crushed rock or screen gravel, graded from $\frac{1}{4}$ in. to 1 in. The water should be pure and about 6 gal. per sack of Portland. Use a mixture of 1 : $1\frac{1}{2}$ : 3, and if possible pour all at once. The inside should be finished in rich mortar rubbed in with a brick. The tank must be kept free of other construction and the ground underneath should be guarded against freezing.

[8] H. C. Boyden, Journal A. S. R. E., January, 1921.

# TABLE 65

## Capacities of Double Pipe Brine Coolers, $1\frac{1}{4}$-In. and 2-In. Pipe, Tons of Refrigeration

| Pipes High | Brine Range | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 2 | | | | | | 3 | | | | | | 5 | | | | | | 10 | | | | | |
| | Mean temperature difference | | | | | | | | | | | | | | | | | | | | | | | |
| | 5 | $7\frac{1}{2}$ | 10 | $12\frac{1}{2}$ | 15 | $17\frac{1}{2}$ | 5 | $7\frac{1}{2}$ | 10 | $12\frac{1}{2}$ | 15 | $17\frac{1}{2}$ | $7\frac{1}{2}$ | 10 | $12\frac{1}{2}$ | 15 | $17\frac{1}{2}$ | 20 | 10 | $12\frac{1}{2}$ | 15 | $17\frac{1}{2}$ | 20 | $22\frac{1}{2}$ |
| | Capacity of cooler 10 ft. long over all | | | | | | | | | | | | | | | | | | | | | | | |
| 2 | ..... | .11 | .33 | .55 | 1.1 | 1.6 | ..... | ..... | .08 | .20 | .48 | .64 | ..... | ..... | ..... | .13 | .16 | .27 | ..... | ..... | ..... | ..... | .11 | .13 |
| 3 | .11 | .45 | 1.0 | 1.4 | 2.2 | 2.8 | ..... | 1.6 | .48 | .90 | 1.3 | 1.8 | ..... | ..... | .27 | .40 | .67 | 1.3 | ..... | ..... | ..... | .20 | .27 | .40 |
| 4 | .28 | 1.0 | 1.6 | 2.3 | 3.1 | 4.4 | .12 | .40 | 1.1 | 1.6 | 2.4 | 2.9 | ..... | .40 | .67 | 1.3 | 1.6 | 2.5 | ..... | ..... | .27 | .53 | .80 | 1.1 |
| 6 | .83 | 2.0 | 3.0 | ..... | ..... | ..... | .48 | 1.2 | 2.2 | 3.2 | 4.4 | ..... | .54 | 1.3 | 2.3 | 3.2 | 3.7 | 5.1 | .27 | .53 | 1.1 | 1.6 | 2.7 | 3.7 |
| 8 | 1.5 | 3.0 | ..... | ..... | ..... | ..... | 1.1 | 2.2 | 3.5 | ..... | ..... | ..... | 1.3 | 2.4 | 3.6 | 4.9 | ..... | ..... | .53 | 1.6 | 2.7 | 4.0 | 4.8 | 6.4 |
| 10 | 2.2 | ..... | ..... | ..... | ..... | ..... | 1.6 | 3.2 | ..... | ..... | ..... | ..... | 2.3 | 3.6 | 5.3 | ..... | ..... | ..... | 1.3 | 2.8 | 4.5 | 5.8 | 7.2 | 8.8 |
| 12 | ..... | ..... | ..... | ..... | ..... | ..... | 2.2 | ..... | ..... | ..... | ..... | ..... | 3.1 | 4.9 | ..... | ..... | ..... | ..... | 2.6 | 4.5 | 6.1 | 8.0 | 9.9 | |
| 14 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 4.0 | ..... | ..... | ..... | ..... | ..... | 4.0 | 5.8 | 8.0 | | | |
| 16 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 5.1 | 7.5 | ... | | | |
| 18 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 6.1 | 8.8 | | | | |
| 20 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 7.5 | | | | | |
| | Gallons per ton per minute, 1.25 specific gravity of brine | | | | | | | | | | | | | | | | | | | | | | | |
| | 18 | | | | | | 12.5 | | | | | | 7.5 | | | | | | 3.75 | | | | | |
| | Capacity of cooler 20 ft. long over all | | | | | | | | | | | | | | | | | | | | | | | |
| 2 | .39 | 1.0 | 1.4 | 2.2 | 3.6 | 5.0 | .12 | .56 | 1.3 | 2.0 | 2.6 | 3.2 | .16 | .40 | .90 | 1.6 | 2.3 | 2.9 | .13 | .26 | .33 | .53 | .80 | 1.3 |
| 3 | 1.1 | 2.2 | 2.5 | ..... | ..... | ..... | .56 | 1.6 | 2.6 | 3.7 | 4.9 | ..... | .67 | 1.6 | 2.6 | 3.6 | 4.6 | 5.6 | .26 | .66 | 1.3 | 2.1 | 3.2 | 4.2 |
| 4 | 1.8 | 3.4 | ..... | ..... | ..... | ..... | 1.2 | 2.6 | 4.1 | ..... | ..... | ..... | 1.6 | 2.9 | 4.2 | 5.6 | 7.3 | ..... | .80 | 1.8 | 3.2 | 4.5 | 5.8 | 7.2 |
| 5 | 2.5 | ..... | ..... | ..... | ..... | ..... | 1.9 | 4.1 | ..... | ..... | ..... | ..... | 2.7 | 4.2 | 6.1 | ..... | ..... | ..... | 1.8 | 3.4 | 5.2 | 6.9 | 8.5 | 10 |
| 6 | ..... | ..... | ..... | ..... | ..... | ..... | 2.6 | ..... | ..... | ..... | ..... | ..... | 3.6 | 5.7 | ..... | ..... | ..... | ..... | 3.2 | 5.1 | 7.2 | 9.3 | 12 | |
| 8 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 5.7 | ..... | ..... | ..... | ..... | ..... | 5.8 | 8.5 | 12 | | | |
| 10 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 8.5 | 12 | | | | |
| 12 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 12 | | | | | |
| 14 | | | | | | | | | | | | | | | | | | | | | | | | |
| 16 | | | | | | | | | | | | | | | | | | | | | | | | |
| 18 | | | | | | | | | | | | | | | | | | | | | | | | |

## TABLE 66

### Capacities of Double Pipe Brine Coolers, 2-In. and 3-In. Pipe, Tons of Refrigeration

| Pipes High | Brine Range | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 2 | | | | | | 3 | | | | | | 5 | | | | | | 10 | | | | | |
| | Mean temperature difference | | | | | | | | | | | | | | | | | | | | | | | |
| | 5 | 7½ | 10 | 12½ | 15 | 17½ | 5 | 7½ | 10 | 12½ | 15 | 17½ | 7½ | 10 | 12½ | 15 | 17½ | 20 | 10 | 12½ | 15 | 17½ | 20 | 22½ |
| | Capacity of cooler 10 ft. long over all | | | | | | | | | | | | | | | | | | | | | | | |
| 2 | .04 | .08 | .16 | .39 | .78 | 1.4 | .01 | .06 | .10 | .16 | .24 | .40 | .01 | .033 | .11 | .13 | .20 | .27 | ..... | ..... | .08 | .13 | .19 | .19 |
| 4 | .14 | .55 | 1.2 | 2.0 | 3.4 | 6.6 | .10 | .20 | .56 | 1.1 | 1.8 | 2.6 | .13 | .20 | .38 | .67 | 1.1 | 1.6 | .05 | .16 | .26 | .33 | .40 | .53 |
| 6 | .67 | 1.7 | 3.3 | 4.9 | 8.0 | ..... | .24 | .96 | 2.0 | 3.3 | 4.8 | 6.4 | .33 | .80 | 1.6 | 2.3 | 3.6 | 5.0 | .26 | .40 | .67 | .94 | 1.6 | 2.1 |
| 8 | 1.3 | 3.1 | 5.4 | 7.9 | ..... | ..... | .64 | 1.9 | 3.7 | 5.4 | 7.7 | 8.4 | .80 | 1.7 | 3.2 | 4.9 | 6.7 | 8.7 | .40 | .80 | 1.6 | 2.4 | 3.5 | 5.1 |
| 10 | 2.2 | 4.4 | 7.9 | ..... | ..... | ..... | 1.2 | 3.2 | 5.6 | 8.2 | ..... | ..... | 1.5 | 3.2 | 5.2 | 7.7 | 10 | 13 | .80 | 1.6 | 2.9 | 4.5 | 6.4 | 8.5 |
| 12 | 3.2 | 6.1 | ..... | ..... | ..... | ..... | 1.9 | 4.6 | 7.6 | ..... | ..... | ..... | 2.4 | 4.8 | 7.7 | 10 | 13 | ..... | 1.3 | 2.9 | 4.8 | 6.9 | 9.6 | 13 |
| 14 | 4.7 | ..... | ..... | ..... | ..... | ..... | 3.0 | 6.7 | ..... | ..... | ..... | ..... | 4.0 | 7.5 | 11 | 14 | ..... | ..... | 2.9 | 5.1 | 8.0 | 12 | 15 | 19 |
| 16 | 5.3 | ..... | ..... | ..... | ..... | ..... | 3.7 | 7.7 | ..... | ..... | ..... | ..... | 4.8 | 10 | 13 | ..... | ..... | ..... | 3.5 | 6.2 | 9.6 | 13 | 17 | 21 |
| 18 | ..... | ..... | ..... | ..... | ..... | ..... | 4.6 | ..... | ..... | ..... | ..... | ..... | 6.3 | ..... | ..... | ..... | ..... | ..... | 4.5 | 8.3 | 13 | 17 | 21 | |
| 20 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 7.7 | ..... | ..... | ..... | ..... | ..... | 6.1 | 11 | 16 | 20 | | |
| | Gallons per ton per minute, 1.25 specific gravity of brine | | | | | | | | | | | | | | | | | | | | | | | |
| | 18 | | | | | | 12.5 | | | | | | 7.5 | | | | | | 3.75 | | | | | |
| | Capacity of cooler 20 ft. long over all | | | | | | | | | | | | | | | | | | | | | | | |
| 2 | .44 | ..... | 1.8 | 2.8 | 3.6 | 5.0 | .12 | .64 | .88 | 1.6 | 2.5 | 3.6 | .16 | .27 | .53 | ..... | 1.7 | 2.4 | .19 | .27 | .32 | .43 | .53 | .80 |
| 4 | 2.5 | 3.9 | 6.3 | ..... | ..... | ..... | .88 | 3.6 | 4.5 | 6.7 | 9.0 | 1 | 1.0 | 2.4 | 4.2 | 6.0 | 8.2 | 10 | .53 | 1.1 | 2.0 | 3.5 | 4.8 | 6.4 |
| 6 | 3.7 | 7.5 | ..... | ..... | ..... | ..... | 2.4 | 5.4 | 8.8 | ..... | ..... | ..... | 3.1 | 6.0 | 8.9 | 13 | 16 | ..... | 1.9 | 3.7 | 6.1 | 8.8 | 12 | 15 |
| 8 | ..... | ..... | ..... | ..... | ..... | ..... | 4.5 | 8.8 | ..... | ..... | ..... | ..... | 6.0 | 10 | 15 | ..... | ..... | ..... | 4.5 | 8.0 | 12 | 16 | 20 | 25 |
| 10 | ..... | ..... | ..... | ..... | ..... | ..... | 6.6 | ..... | ..... | ..... | ..... | ..... | 9.1 | 15 | ..... | ..... | ..... | ..... | 7.7 | 13 | 18 | 23 | 27 | |
| 12 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 13 | ..... | ..... | ..... | ..... | ..... | 12 | 18 | 25 | | | |
| 14 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 15 | 22 | | | | |
| 16 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 18 | | | | | |
| 18 | | | | | | | | | | | | | | | | | | | | | | | | |
| 20 | | | | | | | | | | | | | | | | | | | | | | | | |

## TABLE 67

CAPACITIES OF MULTIPASS SHELL AND TUBE BRINE COOLERS—FLOODED, TONS OF REFRIGERATION PER 24 HOURS AND GALLONS PER MINUTE OF BRINE

| Number | Diameter in Inches | Length in Feet | Velocity of Brine through Cooler, Feet per Minute: 75 — Total brine, gallons per minute | Mean temperature difference of brine and ammonia, degrees F. 10 | 12½ | 15 | 17½ | 20 |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 26 | 6 | 55 | 5.7 | 7.18 | 8.6 | 10.0 | 11.5 |
| 2 | 26 | 9 | 55 | 8.5 | 10.7 | 12.8 | 14.9 | 17.2 |
| 3 | 26 | 12 | 55 | 11.4 | 14.4 | 17.2 | 20.0 | 22.9 |
| 4 | 34 | 9 | 90 | 13.6 | 17.2 | 20.5 | 23.8 | 27.4 |
| 5 | 34 | 12 | 90 | 18.1 | 22.8 | 27.3 | 31.7 | 36.4 |
| 6 | 34 | 18 | 90 | 27.2 | 34.3 | 41.0 | 47.7 | 54.8 |
| 7 | 42 | 12 | 190 | 33.1 | 41.8 | 50.0 | 58.0 | 66.5 |
| 8 | 42 | 18 | 190 | 50.0 | 62.6 | 75.0 | 87.0 | 100.0 |

| Number | Diameter in Inches | Length in Feet | Velocity of Brine through Cooler, Feet per Minute: 200 — Total brine, gallons per minute | Mean temperature difference of brine and ammonia, degrees F. 7½ | 10 | 12½ | 15 | 17½ |
|---|---|---|---|---|---|---|---|---|
| 1 | 26 | 6 | 140 | 7.74 | 10.3 | 12.9 | 15.5 | 18.1 |
| 2 | 26 | 9 | 140 | 11.5 | 15.4 | 19.3 | 23.2 | 26.0 |
| 3 | 26 | 12 | 140 | 15.5 | 20.6 | 25.8 | 31.0 | 36.2 |
| 4 | 34 | 9 | 230 | 18.5 | 24.6 | 30.7 | 37.0 | 43.1 |
| 5 | 34 | 12 | 230 | 24.6 | 32.7 | 41.0 | 49.2 | 57.4 |
| 6 | 34 | 18 | 230 | 37.0 | 50.0 | 62.0 | 74.0 | 86.2 |
| 7 | 42 | 12 | 510 | 45.0 | 59.8 | 74.8 | 90.0 | 105.0 |
| 8 | 42 | 18 | 510 | 67.4 | 89.8 | 112.0 | 135.0 | 157.0 |

| Number | Diameter in Inches | Length in Feet | Velocity of Brine through Cooler, Feet per Minute: 400 — Total brine, gallons per minute | Mean temperature difference of brine and ammonia, degrees F. 5 | 7½ | 10 | 12½ | 15 |
|---|---|---|---|---|---|---|---|---|
| 1 | 26 | 6 | 290 | 7.15 | 10.7 | 14.3 | 17.9 | 21.2 |
| 2 | 26 | 9 | 290 | 10.7 | 16.0 | 21.4 | 26.8 | 32.0 |
| 3 | 26 | 12 | 290 | 14.3 | 21.4 | 28.6 | 35.8 | 42.9 |
| 4 | 34 | 9 | 440 | 17.1 | 25.5 | 34.1 | 42.6 | 51.1 |
| 5 | 34 | 12 | 440 | 22.6 | 33.9 | 45.3 | 56.7 | 68.0 |
| 6 | 34 | 18 | 440 | 34.1 | 51.0 | 68.2 | 85.4 | 101.0 |
| 7 | 42 | 12 | 970 | 41.5 | 62.0 | 83.0 | 103.0 | 123.0 |
| 8 | 42 | 18 | 970 | 62.2 | 93.0 | 124.0 | 156.0 | 187.0 |

| Number | Diameter in Inches | Length in Feet | Velocity of brine through cooler, feet per minute: 150 — Total brine, gallons per minute | Mean temperature difference of brine and ammonia, degrees F. 8.85 | 11.0 | 13.2 | 15.4 | 17.7 |
|---|---|---|---|---|---|---|---|---|
| 1 | 26 | 6 | 110 | 8.85 | 11.0 | 13.2 | 15.4 | 17.7 |
| 2 | 26 | 9 | 110 | 13.2 | 16.5 | 20.0 | 23.0 | 27.0 |
| 3 | 26 | 12 | 110 | 17.7 | 22.0 | 27.0 | 31.0 | 36.0 |
| 4 | 34 | 9 | 180 | 21.1 | 28.0 | 32.0 | 37.0 | 42.0 |
| 5 | 34 | 12 | 180 | 28.0 | 35.0 | 42.0 | 49.0 | 56.0 |
| 6 | 34 | 18 | 180 | 43.0 | 53.0 | 63.0 | 74.0 | 85.0 |
| 7 | 42 | 12 | 380 | 52.0 | 64.0 | 77.0 | 90.0 | 102.0 |
| 8 | 42 | 18 | 380 | 77.0 | 96.0 | 116.0 | 134.0 | 154.0 |

| Number | Diameter in Inches | Length in Feet | Velocity of brine through cooler, feet per minute: 300 — Total brine, gallons per minute | Mean temperature difference of brine and ammonia, degrees F. 9.7 | 13.0 | 16.3 | 19.5 | 22.7 |
|---|---|---|---|---|---|---|---|---|
| 1 | 26 | 6 | 210 | 9.7 | 13.0 | 16.3 | 19.5 | 22.7 |
| 2 | 26 | 9 | 210 | 14.5 | 20.0 | 24.0 | 29.0 | 34.0 |
| 3 | 26 | 12 | 210 | 20.0 | 26.0 | 33.0 | 39.0 | 46.0 |
| 4 | 34 | 9 | 345 | 23.0 | 31.0 | 39.0 | 47.0 | 54.0 |
| 5 | 34 | 12 | 345 | 31.0 | 41.0 | 52.0 | 62.0 | 72.0 |
| 6 | 34 | 18 | 345 | 47.0 | 62.0 | 78.0 | 93.0 | 108.0 |
| 7 | 42 | 12 | 765 | 57.0 | 75.0 | 95.0 | 113.0 | 132.0 |
| 8 | 42 | 18 | 765 | 85.0 | 113.0 | 141.0 | 170 0 | 198.0 |

| Number | Diameter in Inches | Length in Feet | Velocity of brine through cooler, feet per minute: 500 — Total brine, gallons per minute | Mean temperature difference of brine and ammonia, degrees F. 8.0 | 12.0 | 16.0 | 20.0 | 24.0 |
|---|---|---|---|---|---|---|---|---|
| 1 | 26 | 6 | 365 | 8.0 | 12.0 | 16.0 | 20.0 | 24.0 |
| 2 | 26 | 9 | 365 | 12.0 | 18.0 | 24.0 | 30.0 | 36.0 |
| 3 | 26 | 12 | 365 | 16.0 | 24.0 | 32.0 | 41.0 | 48.0 |
| 4 | 34 | 9 | 550 | 19.0 | 29.0 | 39.0 | 48.0 | 57.0 |
| 5 | 34 | 12 | 550 | 26.0 | 38.0 | 51.0 | 64.0 | 75.0 |
| 6 | 34 | 18 | 550 | 38.0 | 58.0 | 77.7 | 96.0 | 113.0 |
| 7 | 42 | 12 | 1200 | 47.0 | 70.0 | 93.0 | 116.0 | 138.0 |
| 8 | 42 | 18 | 1200 | 70.0 | 105.0 | 140.0 | 175.0 | 206.0 |

## TABLE 68

CAPACITIES OF DOUBLE-PIPE BRINE COOLERS, 1¼-IN. AND 2-IN. PIPE, 10 FT. AND 20 FT. LONG OVER ALL, SPECIFICATIONS, WEIGHTS

| Pipes High | Effective Surface | Effective Surface When Submerged | Height over Top Pipe, Inches | Height over All, Inches | Full Weight Pipe, Weight | Extra Heavy Pipe, Weight | Add for Soldering Joints, Dollars |
|---|---|---|---|---|---|---|---|
| | | | 10-ft. coils | | | | |
| 4 | 15.0 | 19.2 | 24 | 33 | 550 | 610 | 12 |
| 6 | 19.5 | 27.8 | 32 | 39 | 750 | 840 | 17 |
| 8 | 26.0 | 38.2 | 40 | 47 | 1000 | 1120 | 22 |
| 10 | 32.5 | 48.0 | 48 | 63 | 1250 | 1400 | 28 |
| 12 | 39.0 | 57.6 | 56 | 71 | 1500 | 1680 | 33 |
| 14 | 45.5 | 67.2 | 64 | 79 | 1750 | 1960 | 39 |
| 16 | 51.5 | 77.0 | 72 | 87 | 2000 | 2220 | 44 |
| 18 | 58.2 | 86.5 | 80 | 95 | 2250 | 2500 | 49 |
| 20 | 64.9 | 96.0 | 88 | 113 | 2500 | 2800 | 54 |
| | | | 20-ft. coils | | | | |
| 4 | 30.4 | 44.4 | 24 | 33 | 720 | 900 | 11 |
| 6 | 45.6 | 66.6 | 32 | 39 | 1080 | 1320 | 17 |
| 8 | 61.0 | 89.0 | 40 | 47 | 1440 | 1760 | 22 |
| 10 | 76.5 | 111.0 | 48 | 63 | 1800 | 2200 | 28 |
| 12 | 91.0 | 133.2 | 56 | 71 | 2130 | 2620 | 33 |
| 14 | 106.5 | 155.5 | 64 | 79 | 2500 | 3000 | 37 |

Gas and liquid valves included with each coil. No headers included.

## TABLE 69

CAPACITIES OF DOUBLE-PIPE BRINE COOLERS, 2-IN. AND 3-IN. PIPE, 10 FT. AND 20-FT. LONG OVER ALL, SPECIFICATIONS, WEIGHTS

| Pipes High | Effective Surface | Effective Surface When Submerged | Height over Top Pipe, Inches | Height over All, Inches | Full Weight Pipe, Weight | Extra Heavy Pipe, Weight | Add for Soldering Joints, Dollars |
|---|---|---|---|---|---|---|---|
| | | | 10-ft. coils | | | | |
| 2 | 10.5 | 14.4 | 12 | 20 | 430 | 570 | 7 |
| 4 | 21.0 | 28.8 | 24 | 32 | 820 | 980 | 12 |
| 6 | 31.5 | 43.2 | 36 | 44 | 1250 | 1450 | 17 |
| 8 | 42.0 | 57.5 | 48 | 56 | 1650 | 1930 | 22 |
| 10 | 52.5 | 72.0 | 60 | 68 | 2000 | 2400 | 28 |
| 12 | 63.0 | 86.5 | 72 | 80 | 2400 | 2880 | 33 |
| 14 | 73.5 | 101.0 | 84 | 92 | 2820 | 3360 | 39 |
| 16 | 84.0 | 115.0 | 96 | 104 | 3220 | 3840 | 44 |
| 18 | 94.2 | 129.0 | 108 | 116 | 3620 | 4320 | 49 |
| 20 | 105.0 | 144.0 | 120 | 128 | 4050 | 4800 | 54 |
| | | | 20-ft. coils | | | | |
| 2 | 23.0 | 31.5 | 12 | 20 | 680 | 845 | 7 |
| 4 | 46.0 | 63.0 | 24 | 32 | 1320 | 1660 | 12 |
| 6 | 69.0 | 94.5 | 36 | 44 | 1960 | 2460 | 17 |
| 8 | 92.0 | 126.0 | 48 | 56 | 2600 | 3300 | 22 |
| 10 | 115.0 | 157.6 | 60 | 68 | 3240 | 4100 | 28 |
| 12 | 138.0 | 189.1 | 72 | 80 | 3880 | 4950 | 33 |
| 14 | 161.0 | 220.5 | 84 | 92 | 4560 | 5760 | 39 |
| 16 | 184.0 | 252.0 | 96 | 104 | 5200 | 6600 | 44 |
| 18 | 207.0 | 283.6 | 108 | 116 | 5840 | 7400 | 49 |
| 20 | 230.0 | 315.2 | 120 | 128 | 6480 | 8200 | 54 |

Gas and liquid valves included with each coil. Add for gas and liquid headers 40 lb. per coil; for brine valves and headers 75 lb. per coil.

## TABLE 70

### Capacities of Multipass Shell and Tube Brine Coolers, Specifications, Weights

| Number | Diameter, Inside, in Inches | Length over Tube Heads, Inches | Number of 2-In. Tubes | Number of Brine Passes | Thickness of Shell, Inches | Thickness of Heads, Inches | Diameter of Brine Connections, Inches | Effective Tube Surface | Weight, Pounds |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 26 | 6 | 60 | 12 | $\frac{1}{4}$ | $\frac{7}{8}$ | $3\frac{1}{2}$ | 172 | 2,750 |
| 2 | 26 | 9 | 60 | 12 | $\frac{1}{4}$ | $\frac{7}{8}$ | $3\frac{1}{2}$ | 257 | 3,400 |
| 3 | 26 | 12 | 60 | 12 | $\frac{1}{4}$ | $\frac{7}{8}$ | $3\frac{1}{2}$ | 344 | 4,050 |
| 4 | 34 | 9 | 96 | 12 | $\frac{5}{16}$ | 1 | 4 | 410 | 5,900 |
| 5 | 34 | 12 | 96 | 12 | $\frac{5}{16}$ | 1 | 4 | 545 | 7,275 |
| 6 | 34 | 18 | 96 | 12 | $\frac{5}{16}$ | 1 | 4 | 820 | 8,650 |
| 7 | 42 | 12 | 176 | 10 | $\frac{3}{8}$ | 1 | 6 | 998 | 10,800 |
| 8 | 42 | 18 | 176 | 10 | $\frac{3}{8}$ | 1 | 6 | 1498 | 14,250 |

Head in feet and corresponding pounds per square inch for given brine velocities (Feet per minute)

| | 75 | | 150 | | 200 | | 300 | | 400 | | 500 | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Ft. | Lb. | Ft. | Lb. | Ft. | Lb. | Ft. | Lb. | Ft. | Lb. | Ft. | Lb. |
| 1 | .56 | .24 | 2.1 | 9.1 | 3.5 | 1.5 | 7.5 | 3.3 | 13.0 | 5.7 | 20.0 | 8.7 |
| 2 | .75 | .32 | 2.8 | 1.2 | 4.6 | 2.0 | 9.8 | 4.3 | 17.0 | 7.4 | 26.0 | 12.0 |
| 3 | 1.0 | .43 | 3.5 | 1.5 | 5.7 | 2.5 | 12.0 | 5.2 | 20.0 | 8.7 | 33.0 | 15.0 |
| 4 | .75 | .32 | 2.8 | 1.2 | 4.6 | 2.0 | 9.8 | 4.3 | 17.0 | 7.4 | 26.0 | 12.0 |
| 5 | 1.0 | .43 | 3.5 | 1.5 | 5.7 | 2.5 | 12.0 | 5.2 | 20.0 | 8.7 | 33.0 | 15.0 |
| 6 | 1.5 | .65 | 4.8 | 2.1 | 8.0 | 3.5 | 17.0 | 7.4 | 28.0 | 12.1 | 45.0 | 20.0 |
| 7 | 1.0 | .43 | 3.3 | 1.4 | 5.6 | 2.4 | 11.0 | 4.8 | 19.0 | 8.2 | 32.0 | 14.0 |
| 8 | 1.4 | .61 | 4.6 | 2.0 | 8.0 | 3.5 | 16.0 | 7.0 | 27.0 | 12.0 | 44.0 | 19.0 |

#### To Use Capacity Table

*Given:* Temperature of room or substance to be cooled = 32 degrees
Tons refrigeration required = 50

*Assume:* Standard Ammonia Suction pressure of 16 lb., corresponding to 0 deg. F. From 5 deg. to 10 deg. temperature difference should be assumed between ammonia and brine leaving cooler and between brine entering cooler and temperature of room. Using 10 deg. for this case gives:

Temperature of brine entering cooler = 22 degrees
Temperature of brine leaving cooler = 10

$$\text{Mean temperature difference of brine and ammonia} = \frac{22 + 10}{2} - 0 = 16 \text{ deg.}$$

Note—Weights include Ammonia Safety Relief valve, Liquid Gage, Pressure Gage, Gas and Liquid Valves and C. I. supports—no Brine Valves included. Cooler Shells are made of flange steel and Tubes are genuine Charcoal Iron Boiler Tubes.

*From Table:* Next select a brine velocity which will be most suitable from all viewpoints, such as first cost of cooler and brine pump, and power required for pumping brine. Assuming 400 ft. per minute brine velocity through cooler; for 50 tons refrigeration under the conditions stated one finds in the table under 15 degrees mean temperature difference that a 34-in. by 9-ft. cooler has a capacity of 51 tons of refrigeration. The head in feet and corresponding pounds per square inch from table above = 17 ft. or 7.4 lb. per square inch to overcome friction or resistance to flow through cooler.

# TABLE 71

## Sodium Chloride—NaCl

## Specific Heat * (Temperature Table)

*Jessup—Properties of Refrigerating Brines. Refrigerating Engineering, December, 1925. By permission of Director of Bureau of Standards*

| Degrees F. | Degrees C. | Specific Heats of Various Per Cent NaCl Solutions at Various Temperatures | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| − 4 | −20 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 0.784 | | |
| +14 | −10 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 0.835 | 0.827 | 0.821 | 0.815 | 0.809 | 0.804 | 0.798 | 0.794 | .789 | 0.784 | 0.779 |
| 32 | 0 | 0.936 | 0.924 | 0.913 | 0.902 | 0.892 | 0.882 | 0.873 | 0.865 | 0.836 | 0.848 | .841 | .834 | .827 | .821 | .815 | .809 | .803 | .798 | .793 | .788 | .783 |
| 50 | +10 | .938 | .927 | .916 | .906 | .896 | .887 | .878 | .869 | .861 | .853 | .846 | .839 | .832 | .825 | .819 | .813 | .807 | .801 | .796 | .791 | .786 |
| 68 | 20 | .940 | .929 | .919 | .909 | .900 | .890 | .882 | .873 | .865 | .857 | .849 | .842 | .835 | .828 | .822 | .815 | .809 | .803 | .798 | .792 | .787 |
| 86 | 30 | .942 | .932 | .922 | .912 | .902 | .893 | .884 | .876 | .868 | .859 | .852 | .844 | .837 | .830 | .823 | .817 | .811 | .804 | .799 | .793 | .788 |

| Per Cent of NaCl.... | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Specific gravity †.... | 1.036 | 1.044 | 1.051 | 1.058 | 1.066 | 1.073 | 1.081 | 1.088 | 1.096 | 1.104 | 1.112 | 1.119 | 1.127 | 1.135 | 1.143 | 1.151 | 1.159 | 1.167 | 1.176 | 1.184 | 1.192 |
| Degrees Baume..... | 5.08 | 6.06 | 7.04 | 8.00 | 8.96 | 9.91 | 10.85 | 11.78 | 12.71 | 13.63 | 14.54 | 15.45 | 16.35 | 17.25 | 18.14 | 19.02 | 19.90 | 20.78 | 21.65 | 22.51 | 23.37 |
| Degrees F., Freezing Pt., Degrees F..... | 26.7 | 25.5 | 24.2 | 22.9 | 21.6 | 20.2 | 18.8 | 17.3 | 15.7 | 14.1 | 12.4 | 10.6 | 8.7 | 6.7 | 4.6 | 2.4 | 0.0 | −2.5 | −5.2 | +1.4 | +13.3 |

### Per Cent of Sodium Chloride and Corresponding Freezing Points (Specific Gravity Table)

| Specific gravity †..... | 1.03 | 1.04 | 1.05 | 1.06 | 1.07 | 1.08 | 1.09 | 1.10 | 1.11 | 1.12 | 1.13 | 1.14 | 1.15 | 1.16 | 1.17 | 1.18 | 1.19 | 1.20 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Per cent NaCl..... | 4.14 | 5.51 | 6.86 | 8.21 | 9.56 | 10.88 | 12.20 | 13.51 | 14.81 | 16.09 | 17.36 | 18.63 | 19.87 | 21.11 | 22.33 | 23.54 | 24.74 | 25.93 |
| Degrees F., Freezing Pt., Degrees F..... | 27.6 | 26.0 | 24.4 | 22.7 | 20.8 | 19.0 | 17.0 | 14.9 | 12.7 | 10.4 | 8.0 | 5.4 | 2.7 | −0.3 | −3.5 | −3.0 | +9.0 | +27.0 |

### Per Cent of Sodium Chloride and Corresponding Freezing Points (Baumé Density Table)

| Degrees Baumé.. | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Per cent NaCl... | 4.92 | 5.94 | 6.96 | 8.00 | 9.04 | 10.10 | 11.16 | 12.23 | 13.31 | 14.40 | 15.50 | 16.61 | 17.72 | 18.84 | 19.97 | 21.11 | 22.26 | 23.41 | 24.57 |
| Degrees F.—F.P. | 26.7 | 25.5 | 24.3 | 22.9 | 21.6 | 20.1 | 18.6 | 16.9 | 15.2 | 13.4 | 11.5 | 9.4 | 7.3 | 5.0 | 2.4 | −0.3 | −3.2 | −5.0 | +8.0 |

* Specific heat expressed in 20 deg. Cal–g. deg. C.

† Specific gravity, based on 60 deg. F. water and 60 deg. F. brine; Degrees Be, Baumé density; degrees F.—F. P., Freezing point.

## TABLE 72

### Calcium Chloride—$CaCl_2$ (Pure)

### Specific Heat * (Temperature Table)

*Jessup—Properties of Refrigerating Brines. Refrigerating Engineering, December, 1925.*

| Degrees F. | Degrees C. | Specific Heats of Various Per Cent Calcium Chloride Solutions at Various Temperatures | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| −40 | −40 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 0.643 | 0.636 | 0.635 | 0.631 |
| −22 | −30 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 0.663 | 0.656 | .649 | .642 | .639 | .635 | |
| − 4 | −20 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 0.704 | 0.695 | 0.686 | 0.678 | .670 | .663 | .656 | .649 | .644 | .638 | |
| 14 | −10 | ..... | ..... | ..... | ..... | ..... | ..... | 0.781 | 0.768 | 0.756 | 0.745 | 0.734 | 0.723 | .713 | .704 | .694 | .686 | .678 | .670 | .663 | .656 | .649 | .643 | |
| 32 | 0 | 0.882 | 0.867 | 0.853 | 0.839 | 0.825 | 0.812 | 0.799 | .787 | .775 | .763 | .752 | .741 | .731 | .721 | .711 | .702 | .693 | .685 | .677 | .669 | .662 | .655 | .648 | |
| 50 | 10 | 0.887 | 0.872 | 0.858 | 0.844 | 0.831 | 0.818 | 0.805 | 0.793 | 0.781 | 0.770 | 0.759 | 0.748 | 0.738 | 0.728 | 0.718 | 0.709 | 0.70 | 0.692 | 0.683 | 0.676 | 0.669 | 0.662 | 0.655 |
| 68 | 20 | .892 | .877 | .863 | .849 | .836 | .823 | .811 | .799 | .787 | .775 | .764 | .754 | .744 | .733 | .724 | .715 | .706 | .698 | .690 | .682 | .675 | .668 | .661 |
| 86 | 30 | .897 | .882 | .868 | .855 | .842 | .829 | .817 | .805 | .792 | .780 | .769 | .759 | .749 | .739 | .729 | .721 | .712 | .704 | .696 | .689 | .682 | .675 | .668 |
| Per cent of $CaCl_2$ | | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 | 26 | 27 | 28 | 29 | 30 |
| Specific gravity † | | 1.069 | 1.078 | 1.087 | 1.096 | 1.105 | 1.114 | 1.124 | 1.133 | 1.143 | 1.152 | 1.162 | 1.172 | 1.182 | 1.192 | 1.202 | 1.212 | 1.223 | 1.233 | 1.244 | 1.254 | 1.265 | 1.276 | 1.287 |
| Degrees Baumé | | 9.33 | 10.45 | 11.57 | 12.68 | 13.78 | 14.88 | 15.96 | 17.04 | 18.12 | 19.18 | 20.24 | 21.28 | 22.32 | 23.35 | 24.38 | 25.38 | 26.41 | 27.41 | 28.41 | 29.41 | 30.39 | 31.37 | 32.34 |
| Freezing point | | 24.2 | 22.8 | 21.4 | 19.8 | 18.2 | 16.3 | 14.4 | 12.2 | 9.9 | 7.4 | 4.7 | 1.9 | −1.0 | −4.0 | −7.3 | −10.6 | −14.1 | −18.0 | −22.0 | −27.0 | −32.0 | −39.0 | −46.0 |

### Per Cent of Calcium Chloride and Corresponding Freezing Points (Specific Gravity Table) *(Jessup)*

| Specific gravity † | 1.07 | 1.08 | 1.09 | 1.10 | 1.11 | 1.12 | 1.13 | 1.14 | 1.15 | 1.16 | 1.17 | 1.18 | 1.19 | 1.20 | 1.21 | 1.22 | 1.23 | 1.24 | 1.25 | 1.26 | 1.27 | 1.28 | 1.29 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Per cent of $CaCl_2$ | 8.14 | 9.26 | 10.36 | 11.45 | 12.53 | 13.60 | 14.66 | 15.71 | 16.75 | 17.78 | 18.80 | 19.81 | 20.81 | 21.79 | 22.78 | 23.74 | 22.70 | 25.65 | 26.59 | 27.52 | 28.44 | 29.35 | 30.26 |
| Freezing point, degrees F. | 24.0 | 22.5 | 20.8 | 19.1 | 17.2 | 15.1 | 30.0 | 10.6 | 8.1 | 5.4 | 2.6 | −0.4 | −3.4 | −6.4 | −9.7 | −13.2 | −16.8 | −20.7 | −25.0 | −30.0 | −35.0 | −41.0 | −48.0 |

### Per Cent of Calcium Chloride and Corresponding Freezing Points (Baumé Density Table)

| Degrees Baumé | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 | 26 | 27 | 28 | 29 | 30 | 31 | 32 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Per cent of $CaCl_2$ | 8.59 | 9.49 | 10.39 | 11.29 | 12.20 | 13.11 | 14.04 | 14.96 | 15.89 | 16.83 | 17.78 | 18.73 | 19.69 | 20.66 | 21.63 | 22.62 | 23.60 | 24.59 | 25.59 | 26.59 | 27.61 | 28.62 | 29.64 |
| Freezing point, degrees F. | 23.4 | 22.1 | 20.8 | 19.3 | 17.8 | 16.1 | 14.3 | 12.3 | 10.1 | 7.8 | 5.3 | 2.7 | −0.1 | −3.0 | −6.1 | −9.3 | −12.7 | −16.4 | −20.4 | −25.0 | −30.0 | −36.0 | −43.0 |

# TABLE 73

## Heat Content, B.t.u. per Pound from 32 Deg. F.

*E. F. Mueller, A. S. R. E. Journal, July, 1919*

| Degrees F. | Specific Gravity | | | | | | Degrees F. | Specific Gravity | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1.18 | 1.20 | 1.22 | 1.24 | 1.26 | 1.28 | | 0.999 | 1.05 | 1.10 | 1.15 | 1.18 | 1.20 | 1.22 | 1.24 | 1.26 | 1.28 |
| −20 | ..... | ..... | ..... | −34.82 | −34.09 | −33.41 | 10 | ..... | ..... | ..... | −16.70 | −16.01 | −15.60 | −15.20 | −14.85 | −14.52 | −14.24 |
| −19 | ..... | ..... | ..... | −34.16 | −33.42 | −32.78 | 11 | ..... | ..... | ..... | −15.95 | −15.28 | −14.89 | −14.52 | −14.18 | −13.87 | −13.60 |
| −18 | ..... | ..... | −34.28 | −33.50 | −32.77 | −32.14 | 12 | ..... | ..... | ..... | −15.19 | −14.56 | −14.19 | −13.83 | −13.51 | −13.21 | −12.95 |
| −17 | ..... | ..... | −33.61 | −32.84 | −32.13 | −31.51 | 13 | ..... | ..... | ..... | −14.44 | −13.84 | −13.48 | −13.14 | −12.83 | −12.55 | −12.31 |
| −16 | ..... | ..... | −32.93 | −32.18 | −31.48 | −30.87 | 14 | ..... | ..... | ..... | −13.68 | −13.11 | −12.78 | −12.45 | −12.16 | −11.90 | −11.66 |
| −15 | ..... | ..... | −32.25 | −31.52 | −30.83 | −30.24 | 15 | ..... | ..... | ..... | −12.93 | −12.39 | −12.07 | −11.76 | −11.49 | −11.24 | −11.02 |
| −14 | ..... | ..... | −31.58 | −30.86 | −30.18 | −29.60 | 16 | ..... | ..... | ..... | −12.17 | −11.66 | −11.36 | −11.08 | −10.82 | −10.58 | −10.37 |
| −13 | ..... | ..... | −30.90 | −30.19 | −29.63 | −28.96 | 17 | ..... | ..... | ..... | −11.41 | −10.94 | −10.66 | −10.38 | −10.14 | −9.99 | −9.73 |
| −12 | ..... | ..... | −30.22 | −29.53 | −28.88 | −28.33 | 18 | ..... | ..... | ..... | −10.65 | −10.21 | −9.95 | −9.70 | −9.47 | −9.26 | −9.08 |
| −11 | ..... | ..... | −29.54 | −28.86 | −28.23 | −27.69 | 19 | ..... | ..... | ..... | −9.89 | −9.48 | −9.24 | −9.01 | −8.80 | −9.60 | −8.93 |
| −10 | ..... | ..... | −28.86 | −28.20 | −27.59 | −27.05 | 20 | ..... | ..... | ..... | −9.14 | −8.76 | −8.53 | −8.32 | −8.12 | −7.94 | −7.79 |
| −9 | ..... | ..... | −28.18 | −27.54 | −26.94 | −26.41 | 21 | ..... | ..... | −9.11 | −8.38 | −8.03 | −7.80 | −7.62 | −7.44 | −7.28 | −7.14 |
| −8 | ..... | −28.21 | −27.50 | −26.87 | −26.28 | −25.78 | 22 | ..... | ..... | −8.28 | −7.62 | −7.30 | −7.11 | −6.93 | −6.77 | −6.62 | −6.49 |
| −7 | ..... | −27.51 | −26.82 | −26.21 | −25.63 | −25.14 | 23 | ..... | ..... | −7.46 | −6.86 | −6.57 | −6.40 | −6.24 | −6.09 | −5.96 | −5.84 |
| −6 | ..... | −26.82 | −26.14 | −25.54 | −24.98 | −24.50 | 24 | ..... | ..... | −6.33 | −6.10 | −5.84 | −5.69 | −6.55 | −5.42 | −5.30 | −5.20 |
| −5 | ..... | −26.12 | −25.46 | −24.88 | −24.33 | −23.86 | 25 | ..... | ..... | −5.80 | −5.34 | −5.12 | −4.98 | −4.86 | −4.74 | −4.64 | −4.55 |
| −4 | ..... | −25.42 | −24.78 | −24.21 | −23.68 | −23.22 | 26 | ..... | ..... | −4.98 | −4.57 | −4.38 | −4.27 | −4.16 | −4.07 | −3.98 | −3.90 |
| −3 | ..... | −24.72 | −24.07 | −23.54 | −23.03 | −22.58 | 27 | ..... | ..... | −4.15 | −3.81 | −3.65 | −3.50 | −3.47 | −3.39 | −3.32 | −3.25 |
| −2 | −24.66 | −24.02 | −23.42 | −22.88 | −22.38 | −21.94 | 28 | ..... | ..... | −3.32 | −3.05 | −2.92 | −2.85 | −2.78 | −2.71 | −2.65 | −2.60 |
| −1 | −23.94 | −23.32 | −22.74 | −22.21 | −21.72 | −21.30 | 29 | ..... | −2.74 | −2.49 | −2.29 | −2.19 | −2.14 | −2.08 | −2.03 | −1.99 | −1.95 |
| 0 | −23.22 | −22.62 | −22.05 | −21.54 | −21.07 | −20.66 | 30 | ..... | −1.83 | −1.66 | −1.53 | −1.46 | −1.43 | −1.39 | −1.36 | −1.33 | −1.30 |
| +1 | −22.50 | −21.92 | −21.37 | −20.88 | −20.42 | −20.02 | 31 | ..... | −0.91 | −0.83 | −0.76 | −0.73 | −0.71 | −0.70 | −0.68 | −0.66 | −0.65 |
| 2 | −21.78 | −21.22 | −20.69 | −20.21 | −19.76 | −19.38 | 32 | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 |
| 3 | −21.06 | −20.52 | −20.09 | −19.54 | −19.11 | −18.74 | 35 | 3.02 | +2.74 | +2.49 | +2.29 | +2.20 | +2.14 | +2.09 | +2.04 | +1.99 | +1.95 |
| 4 | −20.34 | −19.82 | −19.32 | −18.87 | −18.46 | −18.10 | 40 | 8.05 | 7.30 | 6.65 | 6.12 | 5.87 | 5.72 | 5.57 | 5.44 | 5.32 | 5.22 |
| 5 | −19.62 | −19.12 | −18.63 | −18.20 | −17.80 | −17.46 | 35 | 13.07 | 11.87 | 10.82 | 9.96 | 9.55 | 9.30 | 9.07 | 8.86 | 8.66 | 8.49 |
| 6 | −18.90 | −18.41 | −17.95 | −17.53 | −17.15 | −16.81 | 50 | 18.08 | 16.44 | 15.00 | 13.81 | 13.24 | 12.90 | 12.57 | 12.28 | 12.01 | 11.77 |
| 7 | −18.18 | −17.71 | −17.26 | −16.86 | −16.49 | −16.17 | 55 | 23.08 | 21.02 | 19.18 | 17.67 | 16.94 | 16.50 | 16.09 | 15.72 | 15.37 | 15.06 |
| 8 | −17.45 | −17.01 | −16.58 | −16.19 | −15.84 | −15.53 | 60 | 28.08 | 25.60 | 23.38 | 21.54 | 20.65 | 20.12 | 19.61 | 19.16 | 18.73 | 18.36 |
| 9 | −16.73 | −16.36 | −15.89 | −15.52 | −15.18 | −14.88 | 65 | 33.07 | 30.18 | 27.58 | 25.42 | 24.37 | 23.74 | 23.14 | 22.61 | 22.11 | 21.67 |

## Weights

| Degrees F. | Specific Gravity | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 0.999 | 1.05 | 1.10 | 1.15 | 1.18 | 1.20 | 1.22 | 1.24 | 1.26 | 1.28 | 1.05 | 1.10 | 1.15 | 1.18 | 1.20 | 1.22 | 1.24 | 1.26 | 1.28 |
| | Pounds per cubic foot | | | | | | | | | | Pounds per gallon | | | | | | | | |
| −10 | ..... | ..... | ..... | ..... | ..... | ..... | 77.2 | 78.5 | 79.8 | 81.1 | ..... | ..... | ..... | ..... | ..... | 10.32 | 10.49 | 10.66 | 10.84 |
| 0 | ..... | ..... | ..... | ..... | 74.4 | 75.7 | 77.0 | 78.3 | 79.6 | 80.9 | ..... | ..... | ..... | 9.95 | 10.13 | 10.30 | 10.47 | 10.64 | 10.81 |
| 10 | ..... | ..... | ..... | 72.4 | 74.3 | 75.6 | 76.9 | 78.2 | 79.5 | 80.8 | ..... | ..... | 9.68 | 9.93 | 10.11 | 10.28 | 10.45 | 10.62 | 10.79 |
| 20 | ..... | ..... | 69.1 | 72.3 | 74.2 | 75.4 | 76.7 | 78.0 | 79.3 | 80.6 | ..... | 9.23 | 9.66 | 9.92 | 10.09 | 10.26 | 10.43 | 10.60 | 10.77 |
| 30 | ..... | 65.8 | 69.0 | 72.1 | 74.0 | 75.3 | 76.6 | 77.9 | 79.2 | 80.4 | 8.79 | 9.22 | 9.64 | 9.90 | 10.07 | 10.24 | 10.41 | 10.58 | 10.75 |
| 40 | 62.4 | 65.7 | 68.9 | 72.0 | 73.9 | 75.2 | 76.4 | 77.7 | 79.0 | 80.3 | 8.78 | 9.21 | 9.63 | 9.88 | 10.05 | 10.22 | 10.39 | 10.56 | 10.73 |
| 50 | 62.4 | 65.6 | 68.8 | 71.9 | 73.8 | 75.0 | 76.3 | 77.6 | 78.8 | 80.1 | 8.77 | 9.19 | 9.61 | 9.86 | 10.03 | 10.20 | 10.37 | 10.54 | 10.70 |
| 60 | 62.4 | 65.6 | 68.7 | 71.8 | 73.7 | 74.9 | 76.2 | 77.4 | 78.7 | 79.9 | 8.76 | 9.18 | 9.60 | 9.85 | 10.01 | 10.18 | 10.35 | 10.52 | 10.68 |
| 70 | 62.3 | 65.5 | 68.6 | 71.7 | 73.5 | 74.8 | 76.0 | 77.3 | 78.5 | 79.7 | 8.75 | 9.17 | 9.58 | 9.83 | 10.00 | 10.16 | 10.33 | 10.49 | 10.66 |

Specific gravity (60 deg. F.–39 F.). Mass of Unit Volume of 60 deg. Brine. Mass of Unit Volume of 39 deg. Water.

# TABLE 74

## Magnesium Chloride—$MgCl_2$

### Specific Heat* (Temperature Table)

*Jessup—Properties of Refrigerating Brines. Refrigerating Engineering, December, 1925*

| Degrees F. | Degrees C. | Specific Heats of Various Per Cent $MgCl_2$ Solutions at Various Temperatures | | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| −22 | −30 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 0.711 | 0.700 | 0.699 | | | | | | | |
| − 4 | −20 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 0.739 | 0.728 | .717 | .804 | .696 | 0.686 | 0.676 | 0.666 | 0.657 | | | | |
| 14 | −10 | ..... | ..... | ..... | ..... | ..... | ..... | 0.822 | 0.809 | 0.795 | 0.782 | 0.770 | 0.758 | .746 | .735 | .724 | .713 | .702 | .692 | .682 | .673 | .664 | 0.654 | 0.654 | 0.637 | 0.629 |
| 32 | 0 | 0.927 | 0.912 | 0.897 | 0.882 | 0.868 | 0.854 | 0.840 | .827 | .814 | .801 | .788 | .776 | .764 | .752 | .741 | .730 | .719 | .709 | .698 | .689 | .679 | .670 | .660 | .652 | .643 | .635 |
| 50 | 10 | .929 | .914 | .900 | .885 | .872 | .858 | .845 | .832 | .819 | .806 | .794 | .782 | .770 | .758 | .747 | .737 | .726 | .715 | .705 | .695 | .685 | .676 | .666 | .658 | .649 | .641 |
| 68 | 20 | 0.931 | 0.917 | 0.903 | 0.889 | 0.875 | 0.862 | 0.849 | 0.837 | 0.824 | 0.812 | 0.800 | 0.788 | 0.776 | 0.765 | 0.754 | 0.743 | 0.732 | 0.722 | 0.711 | 0.702 | 0.692 | 0.683 | 0.673 | 0.664 | 0.655 | 0.647 |
| 86 | 30 | .933 | .919 | .905 | .892 | .879 | .866 | .854 | .841 | .830 | .817 | .805 | .794 | .783 | .771 | .760 | .750 | .739 | .728 | .718 | .708 | .698 | .689 | .679 | .671 | .661 | .653 |
| Per cent $MgCl_2$ | | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 | 26 | 27 | 28 | 29 | 30 |
| Specific gravity † | | 1.043 | 1.051 | 1.060 | 1.069 | 1.078 | 1.086 | 1.096 | 1.105 | 1.114 | 1.123 | 1.132 | 1.142 | 1.151 | 1.161 | 1.170 | 1.180 | 1.190 | 1.200 | 1.210 | 1.220 | 1.230 | 1.241 | 1.251 | 1.262 | 1.273 | 1.283 |
| Degrees Bé | | 5.91 | 7.06 | 8.20 | 9.32 | 10.44 | 11.53 | 12.64 | 13.73 | 14.82 | 15.88 | 16.94 | 17.99 | 19.03 | 20.08 | 21.11 | 22.13 | 23.14 | 24.17 | 25.16 | 26.16 | 27.15 | 28.14 | 29.12 | 30.09 | 31.06 | 32.02 |
| Degrees F.—F.P. | | 26 4 | 25.0 | 23 5 | 21.8 | 20 0 | 17.9 | 15.7 | 13 1 | 10.3 | 7.3 | 4 0 | 0.4 | −3.5 | −7.7 | −12.2 | −17.3 | −23.0 | −27.0 | −20 0 | −14 0 | −10.0 | −6.0 | −3.0 | −1.0 | +1.0 | +2.0 |

### Per Cent of Magnesium Chloride and Corresponding Freezing Points (Specific Gravity Table) *(Jessup)*

| Specific gravity † | 1.04 | 1.05 | 1.06 | 1.07 | 1.08 | 1.09 | 1.10 | 1.11 | 1.12 | 1.13 | 1.14 | 1.15 | 1.16 | 1.17 | 1.18 | 1.19 | 1.20 | 1.21 | 1.22 | 1.23 | 1.24 | 1.25 | 1.26 | 1.27 | 1.28 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Per cent of $MgCl_2$ | 4.71 | 5.86 | 7.01 | 8.15 | 9.27 | 10.40 | 11 50 | 12 59 | 13.67 | 14.75 | 15.83 | 16.88 | 17.93 | 18.96 | 19.99 | 21.01 | 22.00 | 23.00 | 23.99 | 29.96 | 25.92 | 26.88 | 27.82 | 28.76 | 29.68 |
| Degrees F.—F. P. | 26.7 | 25.2 | 23.5 | 21.5 | 29.4 | 17.0 | 14 4 | 11.5 | 8.3 | 4 8 | 1.0 | −3.0 | −7.4 | −12.0 | −17.2 | −23 1 | −27.0 | −20.0 | −14.0 | −10.0 | −6.0 | −3.0 | −1.0 | +1.0 | +2.0 |

### Per Cent of Magnesium Chloride and Corresponding Freezing Points (Baumé Density Table)

| Degrees, Bé | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 | 26 | 27 | 28 | 29 | 30 | 31 | 32 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Per cent $MgCl_2$ | 5.94 | 6.82 | 7.72 | 8.61 | 9.51 | 10.42 | 11.32 | 12.25 | 13.17 | 14.11 | 15.06 | 16.01 | 16.97 | 17.93 | 18.89 | 19.87 | 20.86 | 21.84 | 22.83 | 23.83 | 24.85 | 25.86 | 26.88 | 27.90 | 28.94 | 29.98 |
| Degrees F.—F.P. | 25 1 | 23.8 | 22.3 | 20.8 | 19.0 | 17.0 | 14.9 | 12.5 | 9.9 | 7.0 | 3.8 | 0.3 | −3.3 | −7.3 | −11.6 | −16.4 | −22.0 | −28.0 | −21.0 | −15.0 | −10.0 | −6.0 | −3.0 | −1.0 | +1.0 | +2.0 |

* Specific heat, expressed in 68 deg. per B.t.u. pound, Deg. F. (20 deg. Cal per g. deg. C.) and in 59 deg. B.t.u. per pound, deg. F. (15 deg. Cal per g., deg. C.).

† Specific gravity, based on 60 deg. F. water and 60 deg. F. brine: degrees Bé. Baumé density; degrees F.—F.P., Freezing point.

## CHAPTER IX

## THE WATER SUPPLY

**Heat removed by the Condenser.**—With the exception of the household refrigerating machine, which is air cooled in the majority of cases, all refrigerating machines use a water cooled condenser in order to absorb the heat which must be removed at the upper temperature in order to liquefy the refrigerant again. The quantity of heat requiring to be removed theoretically is equal to the refrigerating effect plus the work done on the refrigerant by the compressor, provided an expansion cylinder instead of the pressure reducing valve is used. In practice this statement is not correct, because of the use of the irreversible throttling process and because of the heating effect of the compressor which means that more work is done on the gas than would be necessary with non-condensing cylinder walls. As dry compression is used in America almost entirely, the temperature of discharge from the compressor is frequently quite high, as high as 300 deg. F. with certain operating pressures. If the condenser is on the roof, a large amount of this superheat is lost to the atmosphere before the gas reaches the condenser, whereas self-contained units and particularly the shell and tube design of condenser with a short run to the compressor will lose very little superheat before the gas reaches the condenser. The result is that the condensing water has to absorb an amount of heat varying from 220 to 300 B.t.u. per ton of refrigeration per minute.

The amount of water to be used in the plant varies with operating and economic conditions. As already noted (Chapter II), the condenser pressure, the pressure corresponding to the temperature of liquefaction, affects the horse power per ton of refrigeration and also the capacity of the compressor. It is therefore desirable to reduce the condenser pressure as much as it is compatible with economy of operation, whence it is obvious that one may either use a large amount of water that will be raised but a few degrees in temperature or employ a small amount of water to be raised in temperature a relatively large number of degrees. Referring to Table 75,[1] the theoretical values of the amount of water required are

[1] The Electrically Operated Ice Plant, Sloan, Journal of Am. Soc. Refrigerating Engineers.

TABLE 75

GALLONS OF CONDENSER WATER PER TON OF REFRIGERATION

| Condenser Pressure, Pounds per Square Inch, Gage | Corresponding Temperature, Degrees F. | 60 Deg. F. Water | | | | 70 Deg. F. Water | | | | 80 Deg. F. Water | | | | 90 Deg. F. Water | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Range, degrees F. | Water per ton of refrigeration, gallons per minute of suction pressure, pounds gage | | | Range, degrees F. | Water per ton of refrigeration, gallons per minute of suction pressure, pounds gage | | | Range, degrees F. | Water per ton of refrigeration, gallons per minute of suction pressure, pounds gage | | | Range, degrees F. | Water per ton of refrigeration, gallons per minute of suction pressure, pounds gage | | |
| | | | 15 | 20 | 25 | | 15 | 20 | 25 | | 15 | 20 | 25 | | 15 | 20 | 25 |
| 126 | 75 | 10 | 2.90 | 2.85 | 2.80 | | | | | | | | | | | | |
| 131 | 77 | 12 | 2.40 | 2.35 | 2.30 | | | | | | | | | | | | |
| 136 | 79 | 14 | 2.05 | 2.00 | 1.95 | | | | | | | | | | | | |
| 141 | 81 | 16 | 1.85 | 1.80 | 1.75 | 6 | 4.87 | 4.80 | 4.70 | | | | | | | | |
| 147 | 83 | 18 | 1.65 | 1.60 | 1.55 | 8 | 3.67 | 3.60 | 3.55 | | | | | | | | |
| 152 | 85 | 20 | 1.47 | 1.45 | 1.41 | 10 | 2.95 | 2.90 | 2.85 | | | | | | | | |
| 158 | 87 | 22 | 1.35 | 1.32 | 1.30 | 12 | 2.45 | 2.40 | 2.35 | | | | | | | | |
| 163 | 89 | 24 | 1.25 | 1.22 | 1.20 | 14 | 2.10 | 2.05 | 2.00 | | | | | | | | |
| 169 | 91 | 26 | 1.15 | 1.13 | 1.10 | 16 | 1.90 | 1.85 | 1.80 | 6 | 5.00 | 4.90 | 4.80 | | | | |
| 175 | 93 | 28 | 1.08 | 1.05 | 1.03 | 18 | 1.68 | 1.63 | 1.60 | 8 | 3.75 | 3.70 | 3.65 | | | | |
| 181 | 95 | 30 | 1.00 | 0.99 | 0.97 | 20 | 1.50 | 1.48 | 1.45 | 10 | 3.00 | 2.95 | 2.90 | | | | |
| 187 | 97 | 32 | 0.95 | 0.93 | 0.90 | 22 | 1.38 | 1.35 | 1.32 | 12 | 2.52 | 2.48 | 2.43 | | | | |
| 194 | 99 | ..... | ..... | ..... | ..... | 24 | 1.28 | 1.25 | 1.22 | 14 | 2.13 | 2.10 | 2.06 | | | | |
| 200 | 101 | ..... | ..... | ..... | ..... | 26 | 1.17 | 1.15 | 1.13 | 16 | 1.92 | 1.85 | 1.83 | 6 | 5.10 | 5.00 | 4.90 |
| 207 | 103 | ..... | ..... | ..... | ..... | 28 | 1.10 | 1.07 | 1.05 | 18 | 1.70 | 1.67 | 1.63 | 8 | 3.83 | 3.75 | 3.70 |
| 214 | 105 | ..... | ..... | ..... | ..... | 30 | 1.02 | 1.00 | 0.98 | 20 | 1.53 | 1.50 | 1.48 | 10 | 3.07 | 3.00 | 2.95 |
| 221 | 107 | ..... | ..... | ..... | ..... | 32 | 0.95 | 0.93 | 0.92 | 22 | 1.40 | 1.37 | 1.35 | 12 | 2.57 | 2.51 | 2.47 |
| 229 | 109 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 24 | 1.30 | 1.28 | 1.25 | 14 | 2.19 | 2.13 | 2.10 |
| 236 | 111 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 26 | 1.20 | 1.16 | 1.15 | 16 | 1.95 | 1.90 | 1.88 |
| 244 | 113 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 28 | 1.12 | 1.08 | 1.06 | 18 | 1.74 | 1.69 | 1.65 |
| 251 | 115 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 30 | 1.05 | 1.02 | 1.00 | 20 | 1.56 | 1.53 | 1.50 |
| 259 | 117 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 32 | 1.00 | 0.98 | 0.95 | 22 | 1.43 | 1.40 | 1.37 |

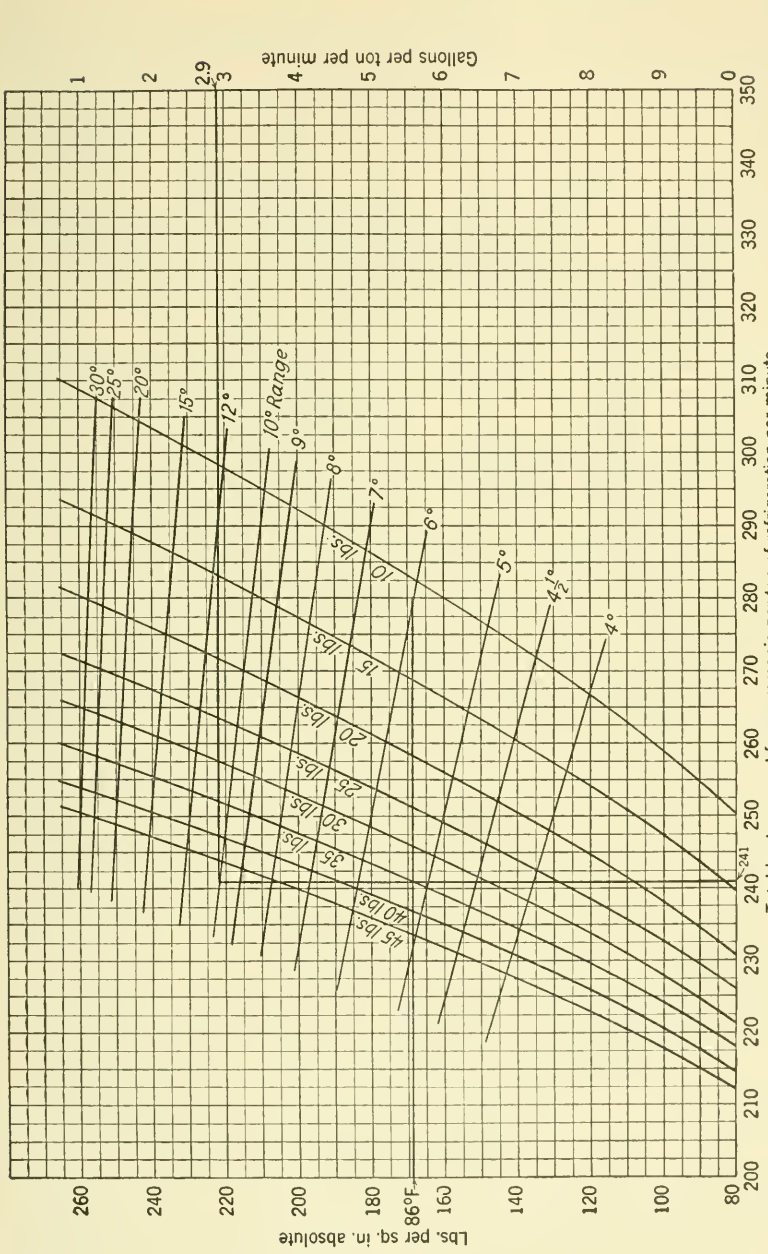



FIG. 182.—Condensing Water Chart.

listed for various initial temperatures and number of degrees rise of temperature of the water. Whether deep well, cooling tower or spray nozzle water is used a certain amount of pumping is required, and the amount of this pumping is justified differently in each type of installation. This point is well brought out by a chart of the heat absorbed by the condenser water, as shown in Fig. 182.

In Fig. 182 the ordinates are condenser pressures, and the abscissa are the heat removed (assuming adiabatic compression and dry saturated gas entering the compressor) per ton of refrigeration per minute. The number of gallons of condensing water can be found also from the scale on the right-hand side of the chart. As an example of the use of the diagram take standard conditions of operation, which are 86 degree condenser and 5 degree evaporator temperatures (corresponding to 154.5 lb. and 19.6 lb. gage, respectively, for ammonia) from which the heat to be removed is found to be 241 B.t.u. per ton per minute. Taking a ten-degree rise of water temperature this condition can be obtained, as regards condenser and suction pressure, with the use of 2.9 gal. of water per minute at an initial temperature of 70 deg. Assuming instead a 5-degree rise and 5.8 gal. per ton of refrigeration per minute, the new condenser pressure will be (still assuming that there is no air or other non-condensible gas present) that corresponding to $70 + 5 + 5 = 80$ deg. F., or 138.3 lb. gage. Therefore, doubling the amount of water has had the effect of reducing the head pressure 16.2 lb. and the horse power per ton of refrigeration (Fig. 182) from about 1.29 to about 1.15. In some cases the use of a large amount of water will be found to be justified, especially if the total cost of pumping, plus the overhead due to the piping and machinery, the sprays or the cooling tower, if such have to be used, is not very large. As a rule when the condenser pressure gets over 175 lb. gage, it is wise to increase the volume of water in order to improve in other ways the operating conditions in the compressor room.

**Temperature of Well and Surface Water.**—The most satisfactory source of condensing water is that of the deep well. Figure 183, taken from the United States Geological Survey statistics,[2] shows the approximate temperature of the water from non-thermal wells at a depth of 30 to 60 feet, and Fig. 184 gives the corresponding temperature of the surface water. In this bulletin it is stated: "The temperature of the water in the ground at any place is in general about the same as the mean *annual temperature* of the *air*. Near the surface the temperature of the water follows the temperature changes of the air; at greater

[2] Water Survey Paper 520-F, by W. D. Collins.

depths the water temperature corresponds to the increase in the earth temperature with increasing depth." Table 76 also gives the approxi-

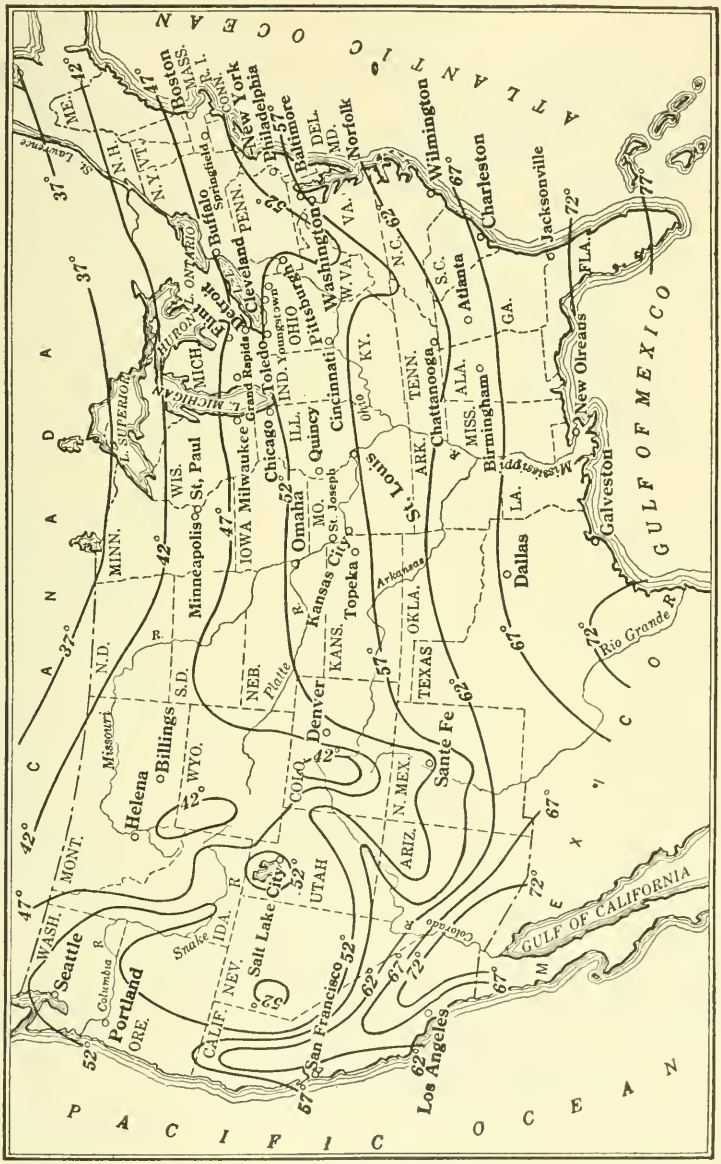



FIG. 183.—Temperature of Non-Thermal Wells.

mate temperature of the surface water and of the air. Very naturally these should be closely related.

**Types of Pumps for Deep Wells.**—There are three methods of pumping deep wells: the reciprocating pump, the Pohlé air lift (Fig. 185), and

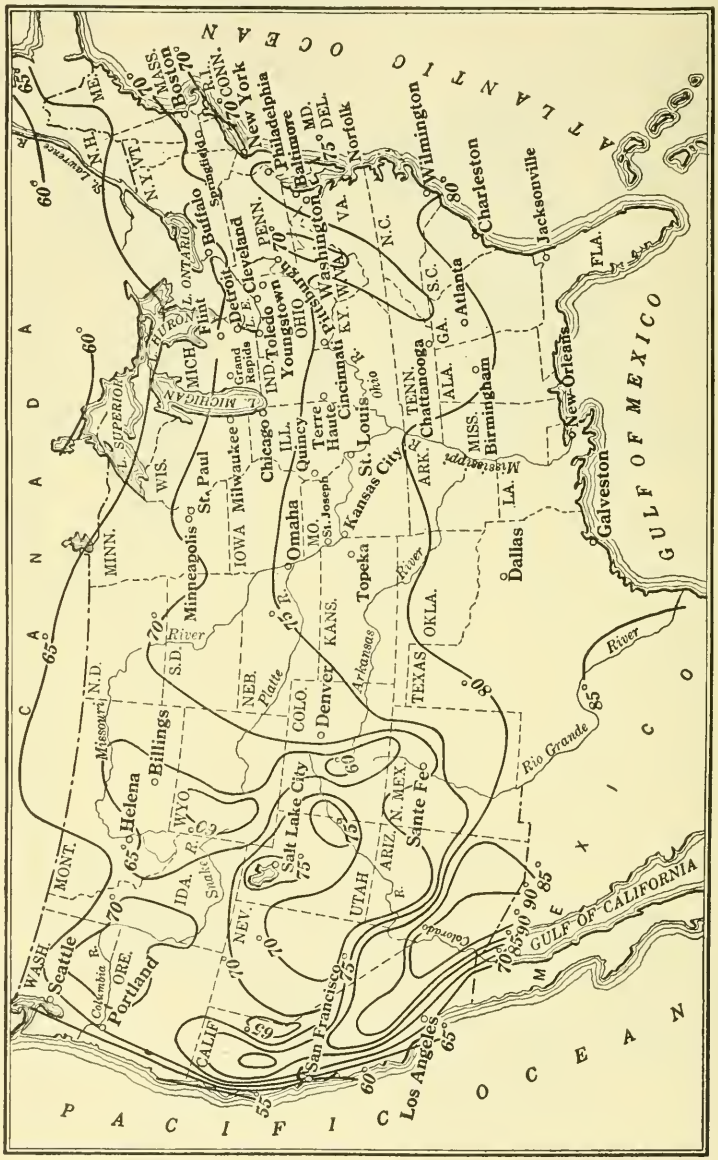



Fig. 184.—Temperature of Surface Water During Summer Months.

the centrifugal pump. The reciprocating pump is best suited to small quantities of water and is usually operated at the high efficiency of

67 to 73 per cent.[3] There is likelihood of trouble, however, due to leaky valves, and the capacity of the well is usually limited to 400 to

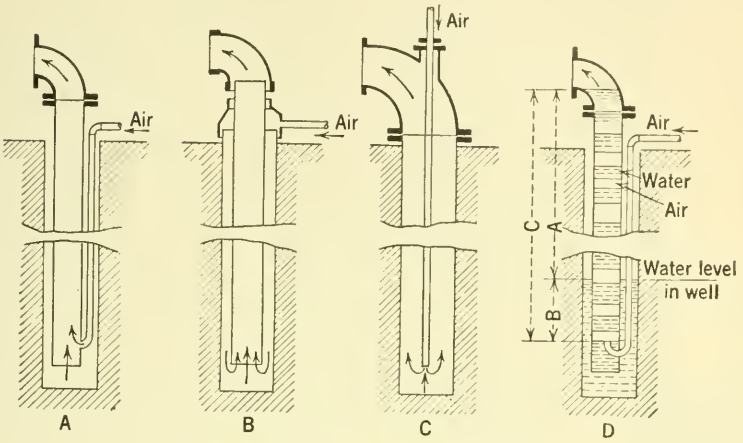



Fig. 185.—The Pohlé Air Lift.

600 gal. per min. so that if larger quantities are required more wells must be sunk.

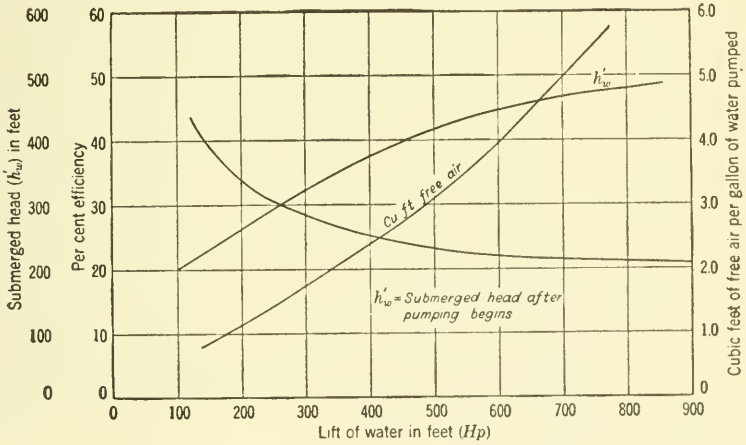



Fig. 186.—Performance of Air Lift Pumps.

The air lift, first proposed by Dr. Julius Pohlé in 1888, can raise much more water from the same sized well than can the plunger pump. Armour & Co., Chicago, were able in 1912 to increase their quantity of

[3] Deep Well Water Pumping, G. B. Mulley, Power, 6-23-25.

# TABLE 76

## Mean Monthly Temperature of Surface Water and Air

W = Water temperatures from waterworks officials.
A = Air temperatures from U. S. Weather Bureau.

| Locality | Year | Mean Monthly Temperature | | | | | | | | | | | | | | | | | | | | | | | | Maximum Daily Temperature of Water | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | January | | February | | March | | April | | May | | June | | July | | August | | September | | October | | November | | December | | | |
| | | W | A | W | A | W | A | W | A | W | A | W | A | W | A | W | A | W | A | W | A | W | A | W | A | July | Aug. |
| 1. Baltimore, Md. | 1918 | 35 | 24 | 34 | 35 | 43 | 47 | 50 | 53 | 65 | 69 | 69 | 71 | 71 | 75 | 74 | 78 | 67 | 65 | 57 | 61 | 50 | 47 | 40 | 42 | 76 | 78 |
| | 1919 | 37 | 38 | 37 | 38 | 45 | 46 | 50 | 53 | 57 | 65 | 67 | 73 | 73 | 78 | 72 | 75 | 68 | 70 | 61 | 63 | 48 | 47 | 39 | 33 | 77 | 76 |
| | 1920 | 35 | 29 | 34 | 33 | 39 | 45 | 51 | 53 | 59 | 61 | 69 | 73 | 74 | 76 | 75 | 75 | 70 | 70 | 62 | 63 | 48 | 47 | 40 | 40 | 77 | 78 |
| | 1921 | 36 | 37 | 39 | 39 | 51 | 55 | 58 | 59 | 61 | 63 | 73 | 75 | 76 | 80 | 73 | 74 | 72 | 74 | 58 | 58 | 46 | 48 | 37 | 37 | 80 | 80 |
| | 1922 | 34 | 33 | 36 | 38 | 47 | 45 | 60 | 56 | 69 | 67 | 73 | 75 | 76 | 77 | 75 | 74 | 71 | 71 | 62 | 61 | 49 | 49 | 38 | 37 | 78 | 77 |
| | 1923 | 34 | 37 | 35 | 32 | 39 | 44 | 48 | 54 | 57 | 64 | 64 | 77 | 66 | 77 | 69 | 75 | 66 | 70 | 61 | 57 | 52 | 46 | 44 | 45 | 68 | 73 |
| 2. Birmingham, Ala. | 1923 | 48 | 52 | 47 | 46 | 51 | 54 | 58 | 62 | 67 | 69 | 72 | 77 | 80 | 78 | 81 | 78 | 78 | 76 | 69 | 64 | 63 | 52 | 53 | 53 | | |
| 3. Chattanooga, Tenn. | 1923 | 49 | 47 | 49 | 42 | 52 | 51 | 58 | 59 | 65 | 65 | 73 | 75 | 77 | 77 | 79 | 77 | 76 | 73 | 69 | 61 | 56 | 50 | 52 | 49 | | |
| 4. Cincinnati, Ohio | 1923 | 39 | 36 | 37 | 28 | 43 | 40 | 52 | 52 | 64 | 61 | 79 | 72 | 81 | 76 | 82 | 74 | 75 | 68 | 64 | 54 | 50 | 44 | 44 | 44 | | |
| 5. Cleveland, Ohio | 1923 | 36 | 30 | 33 | 24 | 36 | 35 | 40 | 46 | 47 | 54 | 59 | 71 | 66 | 71 | 69 | 69 | 68 | 65 | 65 | 52 | 52 | 42 | 43 | 40 | 68 | 72 |
| 6. Detroit, Mich. | 1918 | 32 | 13 | 32 | 24 | 32 | 40 | 37 | 45 | 45 | 62 | 60 | 67 | 67 | 71 | 70 | 75 | 60 | 57 | 55 | 56 | 46 | 42 | 38 | 36 | | |
| | 1919 | 33 | 31 | 32 | 29 | 35 | 36 | 41 | 46 | 51 | 57 | | 74 | 71 | 75 | 70 | 70 | 66 | 67 | 59 | 56 | 44 | 39 | 33 | 23 | | |
| | 1920 | 32 | 17 | | 23 | | 38 | | 41 | 49 | 56 | 64 | 69 | 68 | 70 | 69 | 69 | 66 | 66 | 59 | 59 | 44 | 39 | 37 | 33 | | |
| | 1921 | 32 | 30 | 32 | 30 | 36 | 43 | 44 | 53 | 49 | 62 | | | | | | | | | | | | | | | | |
| 7. Flint, Mich. | 1920 | 34 | 12 | 34 | 19 | 38 | | 45 | | 59 | 56 | 71 | 69 | 71 | 69 | 68 | 69 | 66 | 64 | 60 | 57 | 41 | 36 | 38 | 31 | 79 | 75 |
| | 1921 | 35 | 29 | 35 | 27 | 42 | 40 | 55 | 52 | 69 | 61 | 74 | 71 | 81 | 78 | 74 | 68 | 69 | 67 | 58 | 50 | 46 | 36 | 40 | 28 | 85 | 80 |
| | 1922 | 38 | 20 | 40 | 26 | 40 | 34 | 51 | 47 | 63 | 64 | 74 | 69 | 77 | 71 | 74 | 69 | 68 | 64 | 56 | 51 | 46 | 39 | 35 | 27 | 79 | 79 |
| 8. Grand Rapids, Mich. | 1923 | 34 | 26 | 34 | 20 | 35 | 29 | 47 | 45 | 60 | 57 | 75 | 72 | 76 | 73 | 72 | 68 | 65 | 63 | 54 | 50 | 42 | 40 | 38 | 37 | 78 | 79 |
| 9. Minneapolis, Minn. | 1923 | 34 | 18 | 34 | 10 | 35 | 21 | 42 | 44 | 58 | 60 | 72 | 71 | 76 | 76 | 69 | 67 | 64 | 63 | 54 | 48 | 40 | 39 | 35 | 30 | 81 | 77 |
| 10. New Orleans, La. | 1915 | 43 | 52 | 46 | 56 | 48 | 55 | 57 | 69 | 73 | 77 | 79 | 84 | 82 | 85 | 82 | 83 | 77 | 81 | 71 | 73 | 65 | 66 | 51 | 57 | 84 | 84 |
| | 1916 | 46 | 61 | 45 | 57 | 49 | 64 | 55 | 68 | 66 | 77 | 75 | 82 | 79 | 82 | 83 | 83 | 81 | 79 | 71 | 72 | 61 | 63 | 50 | 57 | 82 | 84 |
| | 1917 | 47 | 60 | 46 | 59 | 54 | 66 | 61 | 68 | 67 | 72 | 77 | 80 | 82 | 83 | 85 | 83 | 81 | 78 | 72 | 66 | 61 | 59 | 54 | 52 | 82 | 87 |
| | 1918 | 48 | 48 | 51 | 63 | 58 | 69 | 63 | 68 | 70 | 76 | 82 | 83 | 83 | 83 | 85 | 82 | 82 | 77 | 73 | 75 | 66 | 62 | 61 | 58 | 84 | 86 |
| | 1919 | 47 | 51 | 46 | 57 | 52 | 64 | 62 | 68 | 67 | 74 | 76 | 80 | 87 | 83 | 89 | 83 | 82 | 80 | 83 | 80 | 67 | 66 | 50 | 57 | 91 | 91 |
| | 1920 | 49 | 56 | 50 | 56 | 51 | 60 | 60 | 69 | 67 | 78 | 77 | 81 | 83 | 82 | 86 | 81 | 84 | 82 | 74 | 71 | 65 | 58 | 54 | 54 | 86 | 88 |
| | 1921 | 53 | 59 | 52 | 60 | 60 | 71 | 64 | 68 | 70 | 74 | 82 | 81 | 88 | 83 | 88 | 84 | 87 | 83 | 74 | 71 | 67 | 67 | 55 | 61 | 90 | 91 |
| | 1922 | 48 | 56 | 46 | 63 | 49 | 62 | 63 | 73 | 71 | 76 | 79 | 82 | 83 | 83 | 84 | 83 | 83 | 81 | 75 | 72 | 66 | 67 | 55 | 64 | 86 | 85 |
| | 1923 | 46 | 61 | 46 | 57 | 50 | 62 | 59 | 70 | 68 | 74 | 77 | 80 | 82 | 80 | 84 | 82 | 81 | 81 | 72 | 70 | 57 | 60 | 52 | 61 | 84 | 86 |
| 11. Pittsburgh, Pa. | 1912 | | 20 | 34 | 25 | 36 | 35 | 48 | 53 | 61 | 64 | 70 | 68 | 75 | 73 | 70 | 69 | 68 | 69 | 57 | 56 | 45 | 44 | 36 | 36 | 81 | 75 |
| | 1913 | 37 | 40 | 34 | 29 | 39 | 43 | 46 | 52 | 61 | 61 | 70 | 70 | 75 | 74 | 77 | 73 | 68 | 65 | 59 | 54 | 46 | 45 | 39 | 36 | 81 | 81 |
| | 1914 | 34 | 34 | 36 | 24 | 37 | 37 | 46 | 49 | 59 | 63 | 75 | 71 | 77 | 74 | 75 | 73 | 68 | 64 | 61 | 58 | 45 | 43 | 37 | 30 | 79 | 77 |
| | 1915 | 34 | 31 | 37 | 37 | 39 | 33 | 48 | 56 | 61 | 58 | 70 | 67 | 72 | 72 | 72 | 69 | 72 | 69 | 57 | 56 | 45 | 45 | 36 | 32 | 77 | 77 |
| | 1916 | 36 | 38 | 36 | 27 | 36 | 34 | 46 | 49 | 59 | 63 | 64 | 65 | 79 | 77 | 79 | 74 | 66 | 64 | 55 | 55 | 43 | 44 | 36 | 33 | 82 | 82 |
| | 1917 | 34 | 32 | 34 | 27 | 37 | 41 | 48 | 50 | 55 | 55 | 66 | 68 | 73 | 74 | 77 | 72 | 66 | 62 | 55 | 49 | 39 | 40 | 34 | 24 | 82 | 84 |
| | 1918 | 34 | 19 | 34 | 33 | 41 | 45 | 50 | 49 | 66 | 68 | 72 | 68 | 75 | 72 | 79 | 77 | 64 | 60 | 55 | 58 | 45 | 44 | 39 | 41 | 82 | 84 |
| | 1919 | 36 | 34 | 34 | 34 | 43 | 42 | 50 | 51 | 55 | 61 | 79 | 75 | 79 | 75 | 75 | 70 | 70 | 66 | 63 | 61 | 46 | 43 | 36 | 28 | 82 | 79 |
| | 1920 | 34 | 24 | 34 | 28 | 37 | 43 | 46 | 47 | 59 | 60 | 70 | 68 | 75 | 70 | 75 | 72 | 70 | 67 | 64 | 60 | 46 | 42 | 37 | 36 | 77 | 77 |
| | 1921 | 32 | 35 | 34 | 36 | 45 | 51 | 57 | 57 | 64 | 62 | 77 | 73 | 82 | 78 | 79 | 70 | 75 | 71 | 57 | 54 | 48 | 45 | 41 | 34 | 86 | 81 |
| | 1922 | 34 | 29 | 34 | 35 | 39 | 43 | 50 | 53 | 64 | 65 | 75 | 71 | 79 | 74 | 75 | 71 | 72 | 70 | 59 | 56 | 46 | 45 | 36 | 36 | 81 | 79 |
| | 1923 | 33 | 33 | 33 | 27 | 37 | 39 | 47 | 50 | 55 | 60 | 76 | 72 | 77 | 73 | 76 | 71 | 72 | 67 | 60 | 52 | 46 | 43 | 42 | 43 | 81 | 81 |
| 12. Pittsburgh region: | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| a | 1923 | 35 | 33 | 38 | 27 | 41 | 39 | 48 | 50 | 60 | 60 | 74 | 72 | 76 | 73 | 75 | 71 | 69 | 67 | 58 | 52 | 44 | 43 | 41 | 43 | | |
| b | 1923 | 40 | 33 | 40 | 27 | 45 | 39 | 52 | 50 | 64 | 60 | 78 | 72 | 79 | 73 | 80 | 71 | 77 | 67 | 71 | 52 | 54 | 43 | 46 | 43 | | |
| c | 1923 | 36 | 33 | 36 | 27 | 38 | 39 | 48 | 50 | 60 | 60 | 74 | 72 | 76 | 73 | 77 | 71 | 72 | 67 | 62 | 52 | 48 | 43 | 42 | 43 | | |
| 13. Quincy, Ill. | 1913 | 34 | 29 | 34 | 25 | 36 | 37 | 51 | 54 | 64 | 65 | 74 | 77 | 80 | 82 | 80 | 83 | 71 | 70 | 58 | 54 | 46 | 49 | 40 | 38 | 82 | 82 |

| | | | | | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1914 | 35 | 34 | 34 | 23 | 37 | 39 | 50 | 54 | 66 | 68 | 77 | 79 | 80 | 82 | 79 | 79 | 70 | 69 | .... | 60 | .... | 45 | .... | 24 | 84 | 81 |
| | 1915 | .... | 24 | .... | 38 | .... | 35 | 63 | 61 | 65 | 62 | 72 | 70 | 78 | 75 | 75 | 71 | 73 | 70 | 57 | 59 | 47 | 47 | 35 | 30 | 81 | 79 |
| | 1916 | 34 | 28 | 34 | 27 | 38 | 41 | 51 | 52 | 62 | 65 | 70 | 70 | 85 | 85 | 83 | 79 | 71 | 67 | 58 | 56 | 44 | 45 | 36 | 27 | 88 | 88 |
| | 1917 | 34 | 28 | 34 | 23 | 38 | 41 | 49 | 49 | 60 | 58 | 70 | 71 | 80 | 79 | 79 | 74 | 72 | 68 | 54 | 47 | 44 | 43 | 35 | 21 | 84 | 84 |
| | 1918 | 34 | 12 | 34 | 29 | 40 | 48 | 51 | 48 | 66 | 68 | 76 | 76 | 77 | 78 | 81 | 82 | 68 | 65 | 62 | 61 | 48 | 46 | 40 | 41 | 81 | 82 |
| | 1919 | 34 | 34 | 36 | 35 | 42 | 46 | 54 | 56 | 61 | 64 | 79 | 78 | 84 | 82 | 81 | 77 | 76 | 74 | 62 | 59 | 45 | 42 | 34 | 26 | 84 | 84 |
| | 1920 | 34 | 26 | 34 | 33 | 39 | 45 | 50 | 50 | 64 | 64 | 78 | 76 | 80 | 78 | 79 | 76 | 75 | 72 | 63 | 64 | 44 | 42 | 37 | 35 | 81 | 81 |
| | 1921 | 34 | 36 | 37 | 40 | 47 | 52 | 55 | 57 | 67 | 69 | 79 | 80 | 84 | 84 | 77 | 79 | 74 | 74 | 58 | 59 | 45 | 43 | 37 | 36 | 86 | 82 |
| | 1922 | 34 | 28 | 34 | 35 | 40 | 45 | 51 | 57 | 67 | 69 | 79 | 78 | 79 | 78 | 78 | 79 | 73 | 73 | 61 | 61 | 49 | 47 | 36 | 34 | 81 | 81 |
| | 1923 | 35 | 37 | 34 | 29 | 37 | 39 | 51 | 54 | 63 | 65 | 78 | 76 | 82 | 81 | 79 | 78 | 70 | 70 | 58 | 54 | 44 | 46 | 39 | 41 | 86 | 84 |
| 14. St. Louis, Mo. | 1921 | .... | .... | .... | .... | .... | .... | 57 | 58 | 68 | 68 | 80 | 78 | 85 | 83 | 80 | 78 | 74 | 74 | 61 | 60 | 49 | 47 | 39 | 39 | 87 | 83 |
| | 1922 | 34 | 30 | 36 | 36 | 44 | 46 | 55 | 58 | 68 | 69 | 78 | 78 | 79 | 79 | 80 | 79 | 74 | 74 | 61 | 62 | 51 | 49 | 38 | 37 | 83 | 83 |
| | 1923 | 38 | 40 | 35 | 30 | 43 | 41 | 51 | 55 | 64 | 64 | 77 | 75 | 81 | 81 | 80 | 78 | 72 | 69 | 60 | 55 | 48 | 48 | 43 | 44 | 85 | 85 |
| 15. St. Joseph, Mo. | 1923 | 34 | 37 | 34 | 29 | 34 | 37 | 48 | 54 | 64 | 62 | 72 | 74 | 78 | 79 | 77 | 76 | 68 | 69 | 61 | 52 | 43 | 45 | 36 | 37 | .... | .... |
| 16. Springfield, Mass. | 1920 | 33 | 19 | 33 | 24 | 33 | 35 | 37 | 45 | 48 | 56 | 59 | 66 | 64 | 70 | 64 | 72 | 59 | 58 | 54 | .... | 42 | .... | 36 | .... | 66 | 66 |
| | 1921 | 33 | 27 | 33 | 30 | 36 | 42 | 44 | 53 | 52 | 58 | 58 | .... | 66 | 74 | 65 | .... | 65 | .... | 54 | 53 | 43 | 40 | 36 | .... | 69 | 68 |
| | 1922 | 33 | .... | 33 | 29 | 34 | 38 | 39 | .... | 51 | .... | 61 | .... | 66 | 74 | 65 | 72 | 61 | 68 | 52 | 57 | 42 | 43 | 35 | 37 | 70 | 67 |
| | 1923 | 33 | 23 | 33 | 23 | 33 | 34 | 39 | 48 | 50 | 57 | 60 | 68 | 65 | 69 | 65 | 68 | 63 | 63 | 52 | 50 | 40 | 39 | 37 | 40 | 66 | 66 |
| 17. Terre Haute, Ind. | 1919 | .... | 34 | .... | 34 | .... | 44 | .... | 54 | .... | 61 | 81 | 76 | 84 | 80 | 81 | 74 | 75 | 71 | 66 | 60 | .... | 42 | .... | 27 | 88 | 86 |
| | 1920 | .... | 24 | .... | 31 | .... | 44 | 52 | 48 | 66 | 63 | 73 | 72 | 77 | 75 | 73 | 73 | 70 | 71 | 61 | 62 | .... | 42 | .... | 34 | 82 | 79 |
| | 1921 | .... | 36 | .... | 38 | .... | 51 | 54 | 57 | 66 | 67 | 79 | 78 | 81 | 81 | 75 | 74 | 70 | 72 | 55 | 57 | .... | 46 | .... | 37 | 86 | 81 |
| | 1922 | .... | 27 | .... | 34 | .... | 44 | 52 | 56 | 63 | 68 | 70 | 75 | 72 | 76 | 73 | 76 | 66 | 72 | 54 | 59 | .... | 46 | .... | 34 | 75 | 77 |
| | 1923 | .... | 36 | .... | 27 | .... | 40 | 48 | 52 | 63 | 62 | 75 | 73 | 72 | 78 | 73 | 74 | 70 | 67 | 54 | 54 | .... | 45 | .... | 43 | 86 | 82 |
| 18. Toledo, Ohio | 1917 | 33 | 26 | 33 | 22 | 40 | 38 | 50 | 45 | 54 | 53 | 69 | 66 | 75 | 73 | 76 | 71 | 67 | 62 | 49 | 46 | 42 | 39 | 34 | 22 | 82 | 84 |
| | 1918 | 34 | 14 | 34 | 28 | 40 | 42 | 49 | 46 | 66 | 65 | 74 | 68 | 74 | 72 | 78 | 76 | 62 | 58 | 56 | 56 | 46 | 43 | 39 | 38 | 79 | 80 |
| | 1919 | 33 | 33 | 32 | 31 | 40 | 37 | 49 | 47 | 58 | 57 | 77 | 74 | 77 | 76 | 73 | 71 | 70 | 68 | 60 | 57 | 44 | 40 | 34 | 24 | 80 | 77 |
| | 1920 | 34 | 18 | 34 | 25 | 39 | 40 | 45 | 42 | 59 | 57 | 71 | 70 | 74 | 71 | 73 | 70 | 70 | 67 | 60 | 61 | 44 | 40 | 38 | 33 | 77 | 77 |
| | 1921 | 34 | 32 | 34 | 32 | 46 | 45 | 54 | 54 | 64 | 62 | 77 | 73 | 82 | 79 | 74 | 71 | 73 | 70 | 56 | 53 | 43 | 40 | 37 | 32 | 88 | 79 |
| | 1922 | 34 | 24 | 34 | 31 | 38 | 38 | 51 | 49 | 65 | 63 | 74 | 70 | 76 | 72 | 75 | 71 | 71 | 68 | 58 | 55 | 46 | 43 | 36 | 30 | 79 | 79 |
| | 1923 | 34 | 30 | 34 | 23 | 36 | 34 | 47 | 47 | 58 | 55 | 76 | 72 | 76 | 73 | 73 | 70 | 68 | 65 | 56 | 51 | 44 | 41 | 39 | 39 | 79 | 79 |
| 19. Topeka, Kans. | 1923 | 42 | 39 | 31 | 30 | 58 | 39 | 52 | 54 | 71 | 62 | 71 | 74 | 82 | 79 | 82 | 77 | 77 | 70 | 68 | 53 | 39 | 46 | 37 | 37 | .... | .... |
| 20. Wilmington, N. C. | 1923 | 49 | 50 | 48 | 46 | 58 | 57 | 63 | 61 | 72 | 68 | 81 | 77 | 84 | 78 | 84 | 79 | 80 | 74 | 68 | 62 | 57 | 53 | 51 | 54 | 86 | 86 |
| 21. Youngstown, Ohio | 1918 | 42 | 15 | 36 | 30 | 46 | 41 | 59 | 48 | 75 | 64 | 83 | 67 | 91 | 70 | 93 | 75 | 84 | 57 | 79 | 55 | 61 | 42 | 48 | 38 | 97 | 102 |
| | 1919 | 45 | 32 | 54 | 32 | 52 | 39 | .... | 48 | 68 | 58 | 91 | 73 | 96 | 74 | 90 | 69 | 86 | 61 | 72 | 59 | 63 | 41 | 50 | 25 | 99 | 95 |
| | 1920 | 50 | 20 | 54 | 25 | 50 | .... | 61 | 44 | 70 | 57 | 81 | 67 | 86 | 69 | 90 | 70 | 90 | 65 | 82 | 58 | 68 | 40 | 52 | 32 | 93 | 95 |
| | 1921 | 50 | 32 | 45 | 32 | 54 | 47 | 63 | 54 | 75 | 61 | 90 | 71 | 93 | 77 | 90 | 69 | 88 | 69 | 75 | 53 | 55 | 42 | 48 | 32 | 99 | 97 |
| | 1922 | 46 | 25 | 45 | 33 | 50 | 40 | 61 | 50 | 75 | 62 | 90 | 69 | 91 | 72 | 93 | 69 | 91 | 66 | 88 | 55 | 77 | 43 | 64 | 32 | 100 | 102 |
| | 1923 | 50 | 30 | 46 | 25 | 48 | 36 | 63 | 47 | 77 | 56 | 93 | 68 | 99 | 70 | 99 | 69 | 97 | 64 | 88 | .... | 68 | 41 | 50 | 41 | 102 | 104 |

1. Gunpowder River impounded for public water supply of Baltimore, Md.
2. Cahaba River impounded for public water supply of Birmingham, Ala.
3. Tennessee River at waterworks intake, Chattanooga, Tenn.
4. Ohio River at waterworks intake, Cincinnati, Ohio.
5. Lake Erie at waterworks intake, Cleveland, Ohio.
6. Detroit River at waterworks intake, Detroit, Mich.
7. Flint River at waterworks intake, Flint, Mich.
8. Grand River at waterworks intake, Grand Rapids, Mich.
9. Mississippi River at waterworks, Minneapolis, Minn.
10. Mississippi River at Carrollton filtration plant of municipal waterworks, New Orleans, La.
11. Allegheny River at Aspinwall filtration plant of municipal waterworks, Pittsburgh, Pa.
12. a, Allegheny River at Colfax, Pa.; b, Monongahela River at Rankin, Pa.; c, Ohio River at Brunot Island. Water temperature furnished by the Philadelphia Co. of Pittsburgh, Pa., from records at plants of the company. Air temperature at Pittsburgh, Pa.
13. Mississippi River at waterworks intake, Quincy, Ill.
14. Mississippi River at waterworks intake, St. Louis, Mo.
15. Missouri River at waterworks, St. Joseph, Mo.
16. Westfield Little River impounded for public water supply of Springfield, Mass.
17. Wabash River at waterworks, Terre Haute, Ind.
18. Maumee River at waterworks, Toledo, Ohio.
19. Kansas River at waterworks, Topeka, Kans.
20. Cape Fear River at waterworks, Wilmington, N. C.
21. Mahoning River at waterworks, Youngstown, Ohio. Temperature of filtered water. Air temperature at Warren, Ohio.

water from 450 to 1200 gal. per min. by substituting an air lift for the plunger pump. In the air lift compressed air is brought into the discharge pipe some distance below the surface of the water in the well, preferably supplied in finely divided bubbles. As the air rises the bubbles act as pistons, and before the flow begins the air serves to lighten the column in the discharge pipe while afterwards the air acts both as a means of lessening the fluid density of the mixture of the air and the water in the discharge pipe and of furnishing motive power required to keep up the continuous flow of water in the discharge pipe. However, the "slip" increases with the size of the air bubbles, and it seems best to supply the air to the discharge pipe through small holes. The overall air lift efficiency is as low as 20 per cent, and the installation has high initial and operating costs but it has the advantage of being able to pump readily even silty and gritty water. Figure 186[4] gives an idea of what can be done with the lift. The diameter of the air pipe which gives the best results has a ratio of 1 : 2 to 1 : $2\frac{1}{2}$ or 3, with the diameter of the discharge pipe, and the velocity of the water at discharge varies from 18 to 26 ft. per sec., whereas the velocity at the bottom is as low as from 6 to 8 ft. per sec. A large pipe means a low velocity and small friction loss, but it means also a much increased slip. The drop of water level when

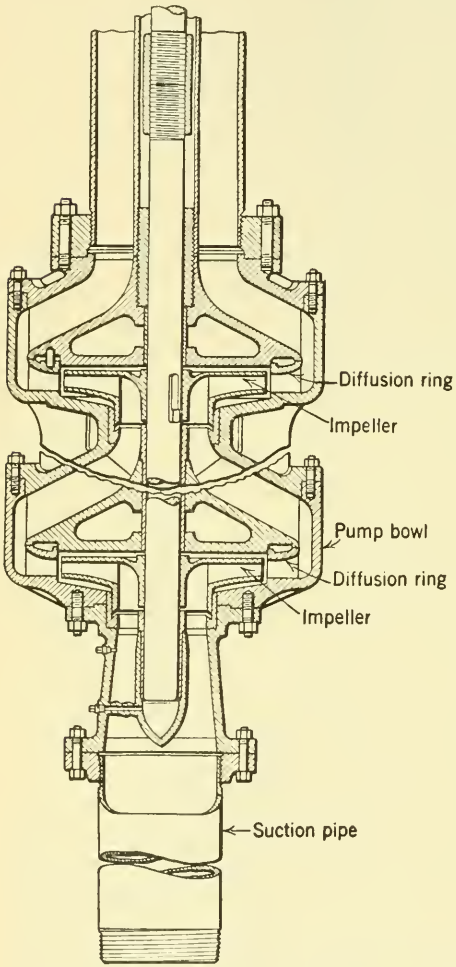



Fig. 187.—The Deep Well Type Centrifugal Pump.

[4] Blaisdell, Power, Nov. 23, 1920.

pumping begins varies, but usually is from 20 to 50 ft. The bottom of the well must be at least 10 to 25 ft. below the air pipe connection to the discharge pipe.

TABLE 77*a*

AIR-LIFT TESTS, POWER-HOUSE WELL

| | Test Number | | | |
|---|---|---|---|---|
| | 1 | 2 | 3 | 4 |
| | Date of test | | | |
| | 2/11/12 | | | 2/18/12 |
| Length of 3-in. air pipe from discharge, feet | 669.0 | 669.0 | 669.0 | 669.0 |
| Surface to water static level, feet | 155.0 | 155.0 | 155.0 | 155.0 |
| Discharge to water static level, feet | 177.0 | 177.0 | 177.0 | 177.0 |
| Air pressure, starting, pounds | 213.0 | 213.0 | 213.0 | 213.0 |
| Air pressure, running, pounds | 153.0 | 153.0 | 153.0 | 153.0 |
| Friction (estimated) air pipe, pounds | 11.9 | 10.0 | 8.6 | 10.2 |
| Running pressure at nozzle, pounds | 141.1 | 143.0 | 144.4 | 141.8 |
| Head on air lift, feet | 343.0 | 338.0 | 335.0 | 342.0 |
| Submergence, running, feet | 326.0 | 331.0 | 334.0 | 327.0 |
| Submergence, running, ratio, per cent | 49.0 | 50.0 | 50.0 | 49.0 |
| Submergence, starting, feet | 492.0 | 492.0 | 492.0 | 492.0 |
| Submergence, starting ratio, per cent | 73.5 | 73.5 | 73.5 | 73.5 |
| Water velocity, feet per second | 5.4 | 5.3 | 5.1 | 5.3 |
| Water temperature, degrees F | 60.0 | 60.0 | 60.0 | 59.0 |
| Drawdown, feet | 166.0 | 161.0 | 158.0 | 165 0. |
| Water pumped, gallons per minute | 1146.0 | 1132.0 | 1109.0 | 1129.0 |
| Free air used, cubic feet per minute | 2031.0 | 1847.0 | 1721.0 | 1880.0 |
| Indicated horsepower, steam end compressor | 383.0 | 380.0 | 326.0 | 360.0 |
| Computed from above | | | | |
| Indicated horsepower hour per 1000 gal. water | 5.58 | 5.15 | 4.90 | 5.31 |
| Specific capacity of well | 6.9 | 7.0 | 7.0 | 6.8 |
| Gallons per minute of water per indicated horsepower | 3.0 | 2.9 | 3.4 | 3.1 |
| Pounds of dry steam per 1000 gal. of water | 84.9 | 78.3 | 80.9 | 80.7 |
| Cubic feet of free air per pound of dry steam | 21.0 | 20.8 | 20.9 | 20.5 |
| Water horsepower | 99.0 | 96.5 | 93.5 | 97.2 |
| Least air horsepower (isothermal compression) | 315.0 | 286.0 | 267.0 | 289.0 |
| Efficiency of air pumping apparatus, per cent | 31.4 | 33.7 | 35.0 | 33.6 |
| Efficiency of compressor and pipe, per cent | 82.4 | 81.7 | 82.0 | 80.4 |
| Efficiency of air lift system, over all, per cent | 25.8 | 27.6 | 28.7 | 27.0 |

**The Centrifugal Pump.**—The centrifugal pump (Fig. 187) is the latest development in deep-well pumping. These are specially designed with small outside diameters with thirteen or more stages. This design of pump has an efficiency of slightly less than three times that of the air lift, and as it has no valves, can handle even water containing silt and slime. The pump is designed for direct connection or belt drive. A summary of tests performed at Armour & Co. is as follows:

TABLE 77*b*

| Date of Test | April, 1921 | March, 1922 | March, 1922 |
|---|---|---|---|
| Static water level | 234.0 | 280.0 | 280.0 |
| Operating water level | 308.0 | 338.0 | 314.5 |
| Drawdown | 54.0 | 58.0 | 34.5 |
| Gallons per minute | 1600.0 | 1470.0 | 1910.0 |
| Total working head | 330.0 | 365.0 | 336.0 |
| I.hp. (input) | 197.5 | 205.6 | 834.0 |
| Gallons per minute per i.hp. | 8.1 | 7.1 | 2.3 |
| Efficiency, overall | 67.5 | 65.9 | 24.7 |
| Air pressure (operating) | ..... | ..... | 156.6 |
| Submergence running (per cent) | ..... | ..... | 46.0 |

**General Water Pumping.**—For general pumping it has been found that, because of the almost universal application of electrical power and because of its own inherent advantages, the direct connected centrifugal pump has replaced the reciprocating pump to a large extent. The centrifugal pump is compact, small sized for its capacity and needs very little attention, and it has an efficiency above 70 per cent at its rated capacities. However, it is not a positive acting machine in the sense of a displacement pump. In its operation water is thrown out by the action of the revolving impeller as a result of which a particular velocity is imparted to the water, and the theoretical head becomes

$$h = \frac{v^2}{2g},$$

where

$h$ = the head in feet;
$v$ = velocity of the water in feet per second.

There are two forms of centrifugal pumps, the volute (Fig. 188) type, which permits the water leaving the impeller to come to partial rest in the spiral casing without the use of guide vanes, and the turbine type,

which has diffuser blades in order to slow up the velocity of the water without impact or shock. The volute type is the more common, and is the better design for variable conditions, while the diffuser blade design will give better economy at the speed and capacity for which it was designed.

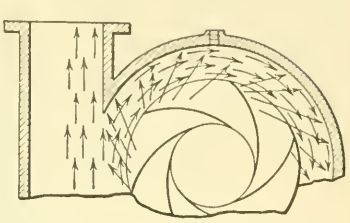


FIG. 188.—Volute Type Centrifugal Pump.

Characteristic curves are shown in Figs. 189 and 190, in which are given the variation of the head pressure with the output in gallons per minute, the efficiency and the required horse power. Figure 191 also

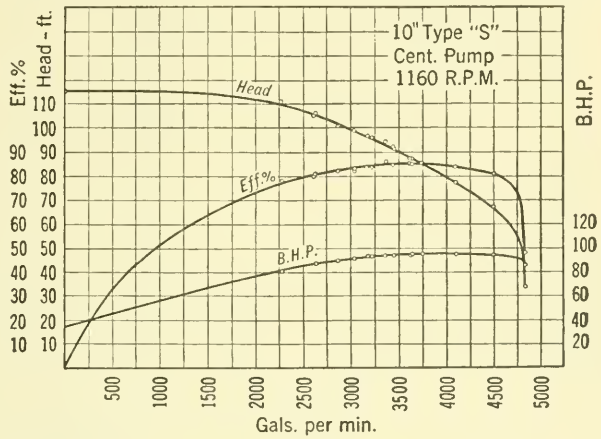



FIG. 189.—Centrifugal Pump Characteristic Curves.

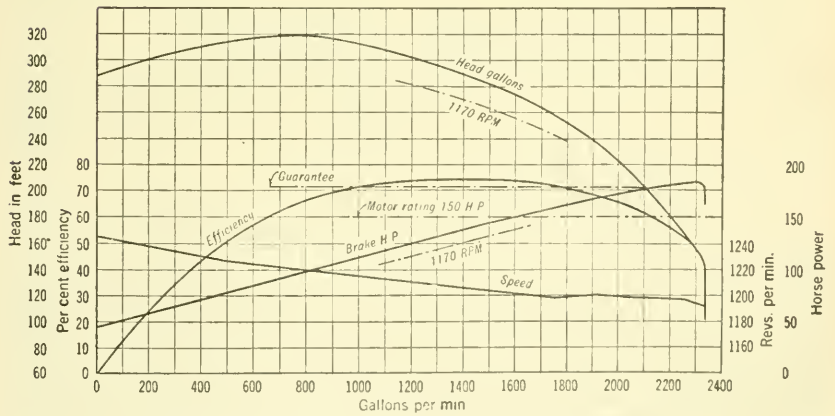



FIG. 190.—Characteristic Curves—Centrifugal Pump.

gives the effect of changing the speed, or the diameter of the impeller, showing that the pressure developed is proportional to some power of the peripheral velocity of the impeller. From experiment it follows that

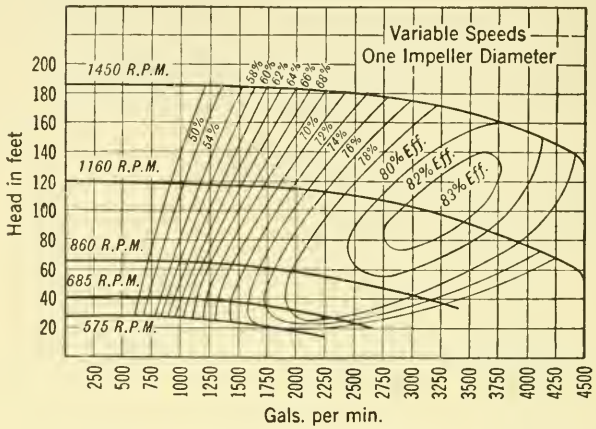



Fig. 191.—Change of Head with Variable Speed—Centrifugal Pumps.

if the diameter of the impeller is constant and the number of revolutions change:

1. The quantity flowing varies as the speed of the pump (the revolution per minute).
2. The head developed varies with the square of the speed.
3. The power required varies as the cube of the speed.

Likewise, if the speed is constant and the diameter of the impeller is changed:

1. The quantity flowing varies as the diameter.
2. The head varies as the square of the diameter.
3. The power required varies as the cube of the diameter.

At times the engineer can vary the speed and the diameter of the impeller so as to get the head and the quantity of liquid pumped to suit best the conditions in the plant.

**The Cooling Tower.**—Where the cost of water is excessive, or where water must be conserved for any reason, advantage may be taken of the cooling tower or the spray method of cooling. The cooling tower as a rule can give a temperature of the water to within three to five degrees of the wet bulb temperature. The spray nozzle can not do so well with a single set of sprays, as usually the maximum range of cooling is from five to eight degrees. In both methods of cooling the principle is to bring the water into intimate contact with air

in order that by the action of evaporation of some of the water into the air the remainder may be cooled.

The cooling tower generally makes use either of a wetted surface with the water flowing by gravity over these surfaces and the air ascending by natural draft, or, simply, the lateral movement of the air through the tower, or of a tower arrangement for securing fine drops of water

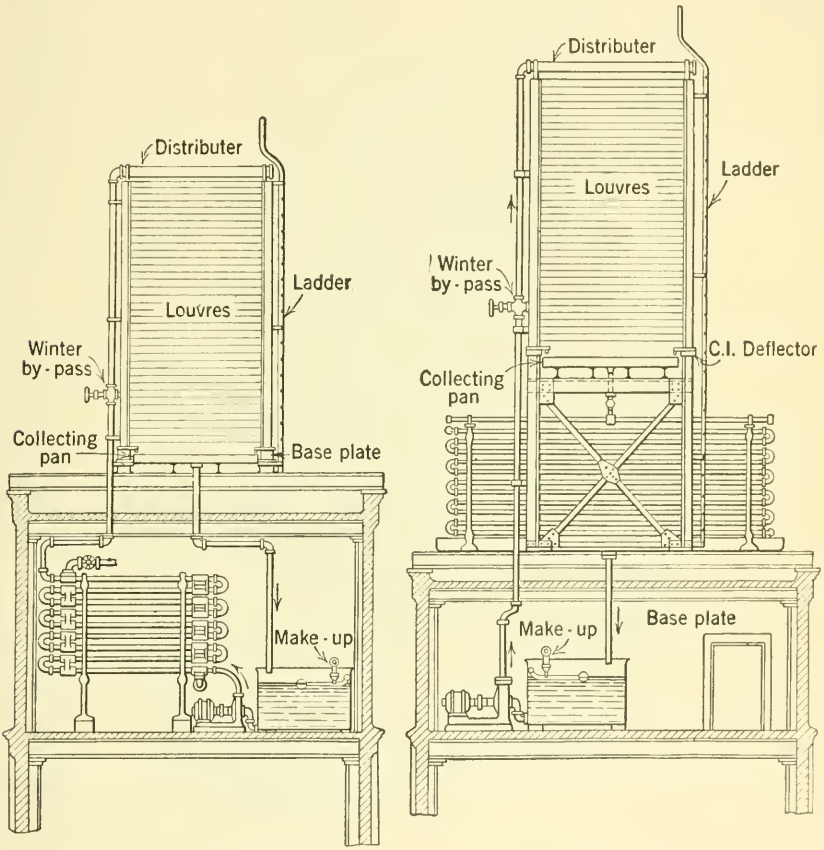



Fig. 192.—Cooling Towers with Ammonia Condensers.

and free fall. This means in refrigeration the placing of the tower in an exposed position in order to get the best possible air flow (Fig. 192). The spray nozzle (Fig. 195) depends on the atomizing of the water by means of a suitable nozzle, using from 5 to 15 lb. per sq. in. water pressure. In either case the cooling of the water is through the ability of the air to evaporate water, the limit of evaporation being the wet

bulb temperature. Some heat can at times be absorbed by the air, but in refrigeration this amount is relatively small because condenser cooling water is seldom much in excess of 100 deg. F. Assuming the latent heat of evaporation to be approximately 1000 B.t.u., the amount of make-up

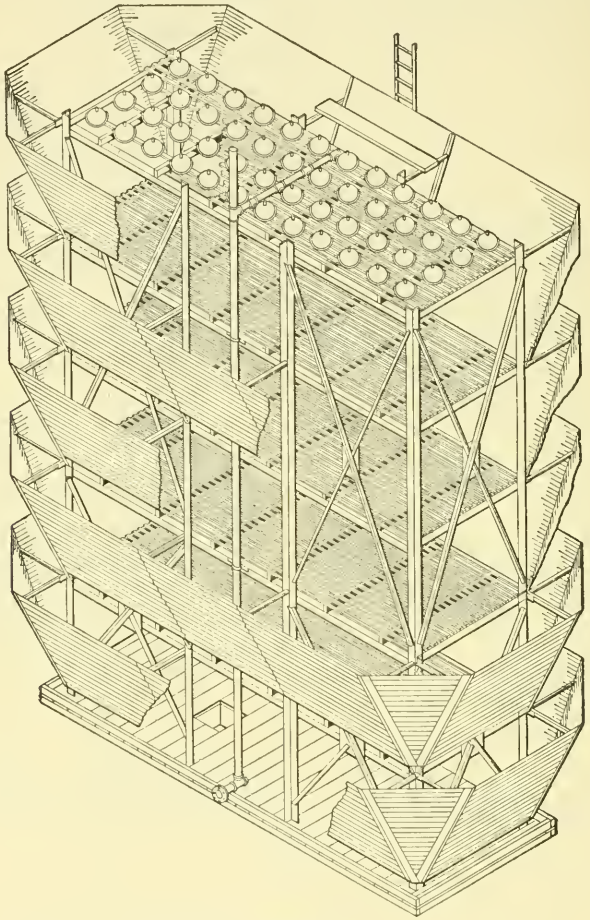


Fig. 193.—Cooling Tower.

water required will be 1.0 per cent for every ten degrees cooling of the water, not allowing for the cooling due to the air.

**Construction.**—The cooling tower is usually constructed with steel frames. The cooling surfaces or decks are made of special anti-corroding metal, structural steel, or of lumber, such as cypress or redwood. The tower is designed with louvres all around, usually designed to catch the

spray caused by the wind and return it to the tower, for unless care is taken in this respect the loss due to windage becomes excessive. The tower is placed sufficiently high as to permit the water to flow by gravity over the condenser if the condenser is of the atmospheric type. The pumping head is that necessary to lift the water up to the dis-

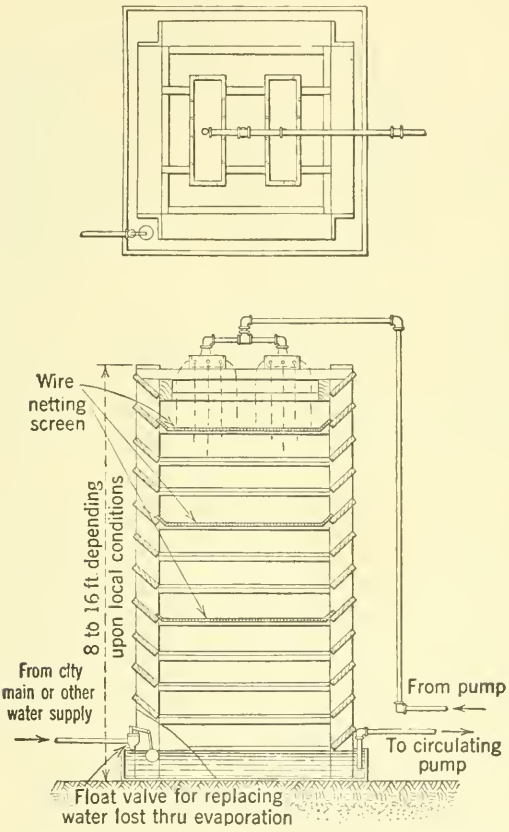



FIG. 194.—Cooling Tower.

tributing troughs. Types of construction are shown in Figs. 192 to 194. If desired, the circulating pump can be placed in the engine room, and if care is taken to make use of the principle of the syphon the only head required on the pump will be that due to the pipe friction, velocity head and the lift from the well to the top of the tower. Table 79 shows the approximate size of the usual tower, and Table 78 gives results of tests on cooling towers.

In the spray nozzle system the pumping head is that necessary to lift

TABLE 78

TYPICAL TEST ON A "BURHORN" COOLING TOWER

| Date | Time | Water Temperature, Degrees | | Air Temperature, Degrees | | Direction of Wind |
|---|---|---|---|---|---|---|
| | | To tower | From tower | Wet bulb | Dry bulb | |
| July 10 | 4:00 p.m. | 87 | 70 | 68 | 84 | N. W. |
| July 11 | 3:30 p.m. | 78 | 68 | 65 | 88 | W. |
| | 4:00 p.m. | 78 | 65 | 64 | 85 | W. |
| | 4:15 p.m. | 74 | 66 | 64 | 84 | W. |
| July 14 | 11:30 a.m. | 73 | 65 | 62 | 78 | N. W. Heavy wind |
| | 1:45 p.m. | 76 | 69 | 67 | 81 | N. W. Wind slight |
| | 4:45 p.m. | 77 | 70 | 67 | 82 | S. W. Slight |
| July 15 | 7:30 a.m. | 75 | 68 | 64 | 71 | W. |
| July 17 | 3:15 p.m. | 79 | 73 | 70 | 81 | S. E. |
| | 4:15 p.m. | 79 | 73 | 70 | 86 | S. E. |

TABLE 79

| Capacity, Gallons per Minute | Dimensions, Inches | | Piping, Inches | | Weight Complete, Pounds |
|---|---|---|---|---|---|
| | Footings | Overall | Supply | Discharge | |
| 8 | ..... | $3\frac{1}{2}\times 7\times 10$ | $\frac{3}{4}$ | $\frac{3}{4}$ | 1,000 |
| 25 | $4\times 4$ | $7\times 7\times 15$ | $1\frac{1}{2}$ | 2 | 2,800 |
| 50 | $4\times 8$ | $7\times 11\times 15$ | 2 | $2\frac{1}{2}$ | 4,600 |
| 75 | $6\times 9$ | $11\times 14\times 20$ | $2\frac{1}{2}$ | 3 | 13,400 |
| 100 | $6\times 12$ | $11\times 17\times 20$ | 3 | $3\frac{1}{2}$ | 15,000 |
| 125 | $6\times 15$ | $11\times 20\times 20$ | $3\frac{1}{2}$ | 4 | 18,000 |
| 150 | $6\times 18$ | $11\times 23\times 20$ | $3\frac{1}{2}$ | 4 | 19,500 |
| 175 | $6\times 21$ | $11\times 26\times 20$ | 4 | $4\frac{1}{2}$ | 21,300 |
| 200 | $13\times 12$ | $19\times 19\times 30$ | 4 | 5 | 32,600 |
| 300 | $13\times 18$ | $19\times 25\times 30$ | $4\frac{1}{2}$ | 6 | 42,000 |
| 400 | $13\times 24$ | $19\times 31\times 30$ | 5 | 6 | 49,900 |
| 500 | $13\times 30$ | $19\times 37\times 30$ | 6 | 7 | 59,700 |
| 600 | $13\times 35$ | $19\times 42\times 30$ | 7 | 8 | 67,700 |
| 700 | $13\times 41$ | $19\times 48\times 30$ | 7 | 8 | 77,500 |
| 800 | $13\times 47$ | $19\times 54\times 30$ | 8 | 10 | 85,200 |
| 900 | $13\times 53$ | $19\times 60\times 30$ | 8 | 10 | 95,900 |
| 1000 | $13\times 59$ | $19\times 66\times 30$ | 10 | 12 | 112,700 |

the water up to the nozzles plus the head necessary to form the spray. The pressure required at the nozzle to form the spray is kept as low as possible, and it should not exceed 10 lb. per sq. in. As the water in spray form will drift excessively in the wind, louvres are essential. Even with the use of louvres the loss becomes very great at times.

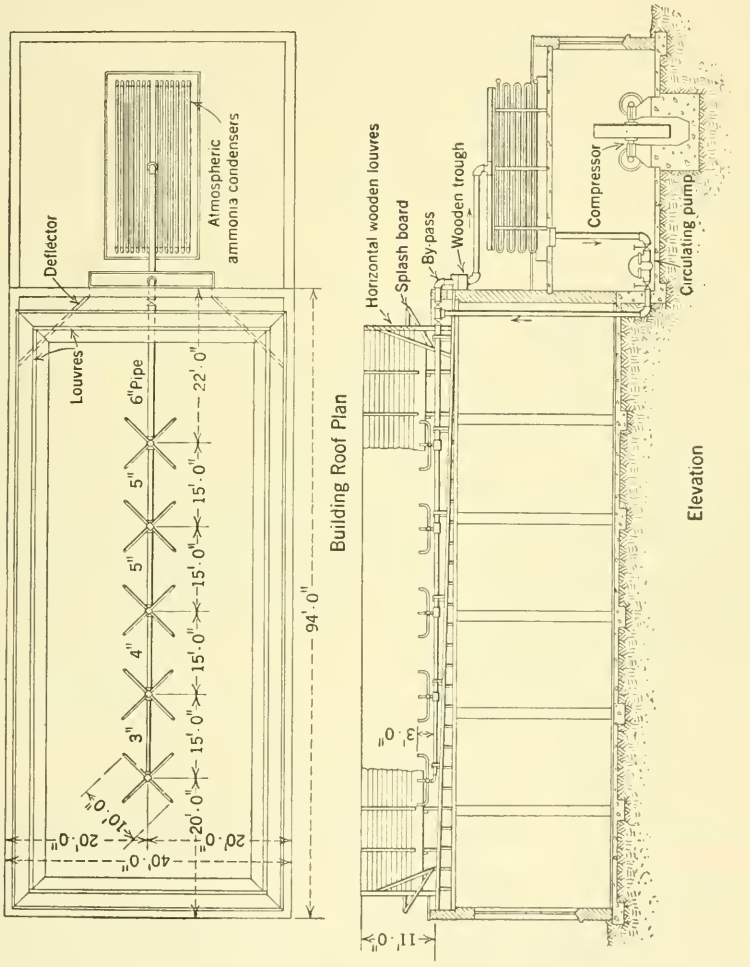



Fig. 195.—Sprays on Roof with Atmospheric Ammonia Condensers.

Figure 195 gives a typical arrangement of sprays on the roof, from which the water flows to the condenser.

**Water Cooling Formula.**—The following equation expresses the sensible heat gained or lost from the air in contact with water:

$$Q_1 = M_a c_p (t_2 - t_1),$$

where

$M_a$ = weight of air, in pounds;
$c_p$ = the specific heat of air;
$t_2$ and $t_1$ = the final and the initial temperatures of the air.

The amount of heat gained or lost from the vapor in contact with the water:

$$Q_2 = \left(\frac{M_1 + M_2}{2}\right)c_p(t_2 - t_1),$$

where

$M_1$ = initial amount of water vapor in the air;
$M_2$ = final amount of vapor.

The amount of heat absorbed during partial evaporation:

$$Q_3 = (V_2 - V_1)r,$$

where

$V_1$ and $V_2$ = the vapor content of the air entering and leaving the system;
$r$ = the latent heat of vaporization.

The amount of the cooling of the water is:[5]

$$Q = M_w(t_2 - t_1),$$

where $M_w$ = the weight of the water.

Coffey and Horne [6] found an expression for the heat flow to and from the wet bulb, which gives a measure of the cooling effect to be found in cooling towers and spray ponds. This expression is:

$$E = (e_T - e_t) + 0.0116(T - t),$$

where

$E$ = measure of the effective cooling head, in inches of mercury;
$T$ = temperature of the water being cooled;
$t$ = temperature of the wet bulb, deg. F.;
$e_T$ = vapor pressure, at temperature $T$, inches of Hg;
$e_t$ = vapor pressure, at temperature $t$, inches of mercury.

Figure 196 shows these relations graphically.

**The Optimum Water Rate.**—In general the refrigerating engineer has not used any scientific method in arriving at the amount of water to be used in the plant. It is generally understood that an increase in the amount of the condensing water means a reduced condenser pressure, and therefore a smaller power consumption by the compressor. How-

[5] Water Cooling System Efficiency, V. J. Azbe, Mechanical Engineering, 1924.
[6] Coffey and Horne, A. S. R. E. Journal, Vol. I, No. 1.

ever condenser water has usually either to be pumped from a deep well, into a cooling tower or through sprays. As the amount of water used by the condensers increases the pumping cost also increases. Doubling the amount of water so as to operate with a 6 deg. F. instead of a 12 deg. F. range of temperature may reduce the condenser pressure from 158.0 to 139.2 lb. per sq. in. Whether this increase in the amount of water is justifiable will certainly depend on the total cost of pumping the water. In all likelihood the engineer will be inclined to set on a certain pressure as the operating maximum, say 185 to 190 lb., and when

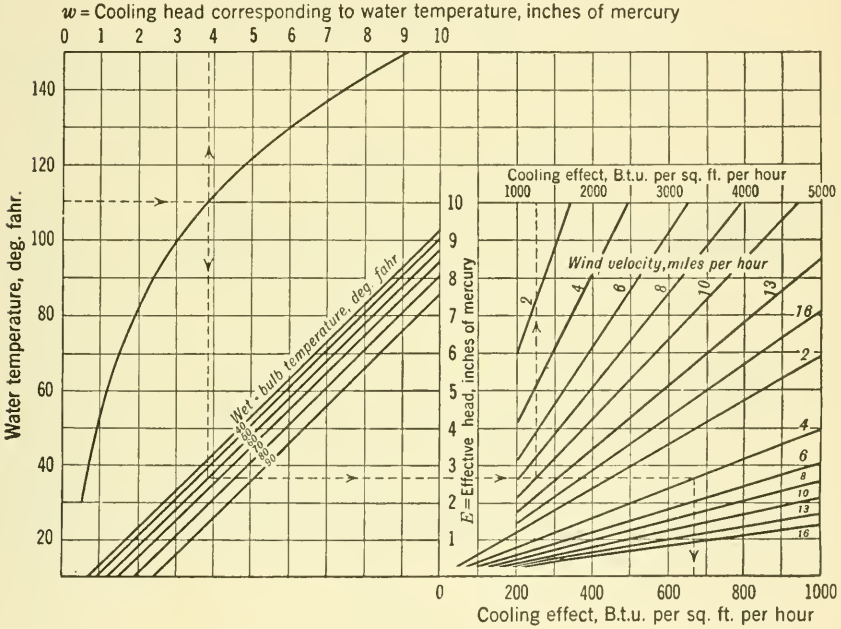



Fig. 196.—Graphical Method of Securing the Cooling Effect.

the condenser pressure becomes greater than this he will attempt to pump more water.

In the following some attempt is made to show how an economical water rate may be determined if certain operating costs are known:[6a]

**Purchased Water.**—The total cost of purchased water is

$$C = \frac{Amt_d}{E_t} + \frac{60BH_c}{1000 \times 8.33t_d}$$

[6a] T. K. Sherwood, Economic Balances in the Design and Operation of the Ammonia Compressor, Refrigerating Engineering, Feb., 1927. R. W. Waterfill, Balancing Compressor Power and Condenser Water Costs, Refrigerating Engineering, Sept., 1927.

where $A$ = the cost of power per hp.-hr.
$m$ = hp. increase per ton of refrigeration per degree increase of condenser temperature
$t_d$ = temperature difference of the condenser water in and out of the condenser
$E_t$ = Combined efficiency of the compressor and motor
$B$ = Cost per 1000 gal. of condensing water
$H_c$ = Heat to be removed from the condenser per ton per minute $= 200 + 42.4W_c$
$W_c$ = The condenser i.hp. per ton of refrigeration
$G = H_c \div 8.33_d$ = the condensing water per ton.

In order to find the minimum cost the first derivative is equated to zero, and therefore

$$\frac{dC}{dt} = \frac{Am}{E_t} - \frac{0.06BH_c}{8.33t_d^2} = 0$$

from which the values for $t_d$ and $G$ may be obtained.

**Pumped Water.**—The total cost of pumped water is

$$C = \frac{mt_d}{E_t} + \frac{8.33G(F_s + f)}{33{,}000E_p}$$

Where $E_p$ is the combined pump and motor efficiency
$F_s$ is the static head on the pump.
$f$ is the friction head due to the flow of the water in the pipes.

In order to make use of the equation $f$ must be replaced by an expression in terms of the variable. Sherwood uses the expression, which is adopted by Waterfill,

$$f = K\left(\frac{D}{t_d}\right)^2$$

where $K$ is a constant giving the value of $f$ for an assumed temperature difference $D$

therefore

$$C = \frac{mt_d}{E_t} + \frac{H_c\left(F_s + K\left(\frac{D}{t_d}\right)^2\right)}{t_dE_p \times 33{,}000}$$

$$\frac{dC}{dt_d} = \frac{m}{Et} - \frac{H_cF_s}{t^2{}_dE_p \times 33{,}000} - \frac{3H_cKD^2}{t^4{}_dE_p \times 33{,}000} = 0$$

**Comparison of Cooling Towers and Sprays.**—The first cost of the spray system should be less than that of the cooling tower, but the loss of water due to drift will be greater with sprays. As a rule the tower

will have the advantage in congested regions where the local conditions are such that an elevated position, free from the wind protection of tall buildings, can be secured. The water in either case must be divided into small particles and the fall, especially in the case of the tower, must be interrupted frequently in order to keep the water in the air for as long a time as possible, whereas the air must be permitted free passage through the tower or over the spray pond.

TABLE 80

List of Nozzle Sizes

| Nozzle Number | Pipe Size for Which Nozzle Is Threaded, Inches | Pressure in Pounds per Square Inch | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 5 | 6 | 7 | 8 | 9 | 10 | 12 | 15 | 20 | 25 | 30 | 40 |
| | | Capacity in U. S. gallons per minute | | | | | | | | | | | |
| 60 | 2 | 53.0 | 58.0 | 62.5 | 67.0 | 71.0 | 74.5 | 82.0 | 92.0 | 106.0 | 118.0 | 130.0 | 150.0 |
| 50 | 2 | 44.0 | 48.0 | 52.0 | 55.5 | 59.0 | 62.0 | 68.0 | 76.0 | 88.0 | 98.0 | 107.0 | 124.0 |
| 45 | 2 | 40.0 | 43.5 | 47.0 | 50.0 | 53.0 | 56.0 | 61.0 | 69.0 | 79.0 | 88.5 | 97.0 | 112.0 |
| 40 | 2 | 35.4 | 38.8 | 42.0 | 45.0 | 47.5 | 50.0 | 55.0 | 61.0 | 71.0 | 79.0 | 87.0 | 100.0 |
| 35 | 2 | 30.8 | 33.8 | 36.4 | 39.0 | 41.2 | 43.5 | 48.0 | 53.5 | 61.5 | 68.0 | 75.0 | 87.0 |
| 30 | 1½ | 26.2 | 28.6 | 31.0 | 33.0 | 35.0 | 37.0 | 40.5 | 45.5 | 52.0 | 58.5 | 64.0 | 74.0 |
| 25 | 1½ | 22.0 | 24.0 | 26.0 | 27.8 | 29.4 | 31.0 | 34.0 | 38.0 | 43.8 | 49.0 | 53.5 | 62.0 |
| 20 | 1½ | 17.5 | 19.4 | 20.8 | 22.2 | 23.5 | 24.8 | 26.0 | 30.5 | 35.0 | 39.0 | 43.0 | 49.5 |
| 15 | 1½ | 13.0 | 14.3 | 15.5 | 16.5 | 17.5 | 18.5 | 20.2 | 22.6 | 26.0 | 29.0 | 32.0 | 37.0 |
| 10 | 1½ | 8.7 | 9.6 | 10.4 | 11.1 | 11.7 | 12.4 | 13.5 | 15.2 | 17.5 | 19.5 | 21.5 | 24.8 |
| 8 | 1 | 7.0 | 7.7 | 8.3 | 8.9 | 9.4 | 9.9 | 10.8 | 12.1 | 14.0 | 15.6 | 17.0 | 19.7 |
| 5 | 1 | 4.4 | 4.8 | 5.2 | 5.5 | 5.9 | 6.2 | 6.8 | 7.6 | 8.8 | 9.8 | 10.8 | 12.4 |
| 4 | ¾ | 3.5 | 3.8 | 4.1 | 4.4 | 4.7 | 4.9 | 5.4 | 6.1 | 7.0 | 7.8 | 8.6 | 9.9 |
| 3 | ½ | 2.6 | 2.9 | 3.1 | 3.3 | 3.6 | 3.7 | 4.1 | 4.5 | 5.2 | 5.9 | 6.4 | 7.4 |
| 2 | ½ | 1.7 | 1.9 | 2.1 | 2.2 | 2.3 | 2.5 | 2.7 | 3.0 | 3.5 | 3.9 | 4.3 | 4.9 |
| A | ⅜ | 1.3 | 1.45 | 1.55 | 1.65 | 1.78 | 1.86 | 2.0 | 2.3 | 2.6 | 2.9 | 3.2 | 3.7 |
| B | ⅜ | .95 | 1.04 | 1.12 | 1.20 | 1.27 | 1.35 | 1.48 | 1.65 | 1.9 | 2.1 | 2.3 | 2.7 |

## CHAPTER X

## ERECTION AND OPERATION

The refrigerating plant is subject to operating difficulties more than is the case with other comparable systems, because of inherent features in the cycle. In consequence of these difficulties, resulting in improper charging and operation, in an easy loss of capacity from a number of causes, in excessive condenser pressure, etc., it is desirable to consider all of these subjects in detail.

**Foundations.**—Foundations for refrigerating machines are required for two reasons—to make the compressor secure, and to absorb the vibrations developed. As considerable care is now being taken in the matter of balancing, vibration is no longer so important a factor as formerly. In this respect the twin vertical single-acting compressor is better than the horizontal double-acting or the duplex compressor. The necessary weight of the foundation being such a variable, it has to be worked out in each case by the individual designers. These foundation weights are made up for fair ground conditions. If the "soil" is rock or compact hard pan it is evident that the size and the weight of the foundation can be decreased, while if it is soft, wet clay or marshy soil, the footings under the foundation must be widened.

**Template and Anchor Bolts.**—A good mixture of concrete should be used, either a $1 : 2\frac{1}{2} : 5$ or a $1 : 2 : 4$ mixture by volume of cement, clean sand and crushed stone. Sufficient time should be allowed for hardening before any weight is placed on the foundation, usually 36 to 48 hours or longer being required. Before pouring, a templet of the anchor bolts with the bolts and anchor plates attached should be set in place and rigidly fastened so that they will not float out of position. Care must be taken to insure the proper level, and the proper alignment with the building wall. The larger machines usually have pockets in the foundation so that access may be had to the anchor plates at any time. In any case ferrules, made of tin, old pipe or thin lumber are so placed that the anchor bolt will hang central and will have a clearance of $\frac{3}{4}$ to 1 in. all around it. While the concrete is green these ferrules can be removed. The anchor plates are screwed up to the middle of the threads on the lower part of the bolt. The pouring of the concrete should

stop at a point from $\frac{1}{2}$ to 1 in. below the base level in order to permit grouting under the machine.

**Leveling and Grouting.**—When the foundation is sufficiently hard to take the weight, the compressor may be moved into place and leveled. Wedges made of wood or iron should be placed near the anchor bolts, and the machine should be leveled in both directions. In a horizontal machine the position of the center of the shaft is required accurately, and this can be located at right angles to the compressor line of centers by means of a triangular templet made with distances between the apexes in the ratio of 3 : 4 : 5. The anchor bolt holes should be filled first with a thin grout of sand and cement in the ratio of 1 to 1 by volume. Then fill in underneath the machine with grouting thin enough to float but not to show water on top, making sure that the grout is worked in under all parts of the compressor bed. In a few hours the wedges can be removed and the foundation finished with a smooth finished surface. The foundation bolts must not be tightened until the grout is thoroughly dried (after two to three days, as a rule). Small compressors can be erected sometimes by drilling out the concrete and using an expansion bolt. These holes are filled with grouting as before.

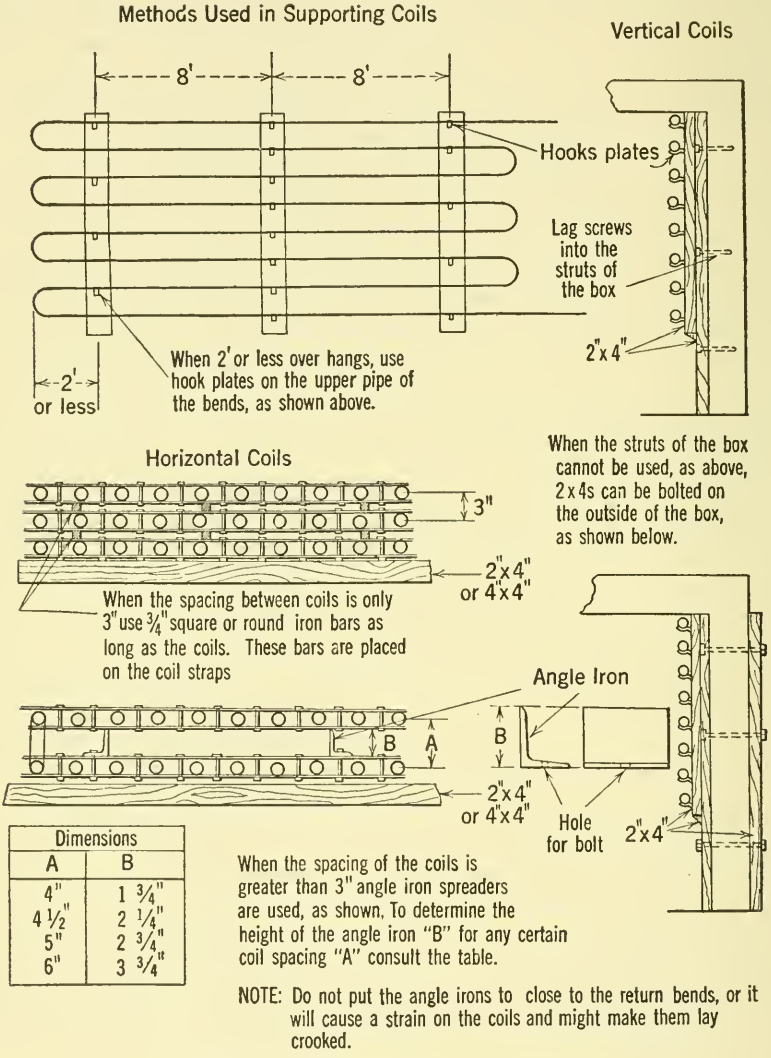
**Piping.**—Piping[1] is one of the important elements in refrigeration machinery. Formerly wrought iron was used extensively, and then, later, extra heavy pipe was specified. There is no standard practice at the present, but the use of full weight black steel pipe is quite common, even for condensers and the line from the compressor to the condenser. Yet some concerns continue to specify the use of extra heavy pipe for the evaporating coils in the tank. In general, the feeling prevails that steel pipe can be relied on much better now than, (say), 10 years ago, and that care in the erection and prevention of excessive corrosion have changed conditions so that extra heavy pipe is no longer justified.

The essential of good pipe work is clean, accurately cut pipe threads, using sharp dies with Briggs standard threads for work in the United States. These threads should be cleaned thoroughly before making up by brushing them with gasoline and wiping clean the pipe threads and the flanges or screwed fittings. Making up the joint so that it will be *hot* is no indication that it will be tight, as the heat indicates friction only, which, of course, could be developed by dirt or scale as well as by tightness.

**Kind of Joint.**—Both soldered and lithage joints are used, although the tendency appears to be towards the elimination of the former except in condenser practice where the soldering is done at the factory. The lithage joint is as easy to make up as a first-class steam fitting connection.

[1] See Chapter IV.

The solder joint is made by cleaning the fittings and the pipe until the metal is bright. The pipe and the fittings are then dropped into molten solder, made of half-and-half of tin and lead, and the threads


Methods Used in Supporting Coils
Vertical Coils
8'
8'
Hooks plates
Lag screws into the struts of the box
2"x4"
2' or less
When 2' or less over hangs, use hook plates on the upper pipe of the bends, as shown above.
When the struts of the box cannot be used, as above, 2x4s can be bolted on the outside of the box, as shown below.
Horizontal Coils
3"
2"x4" or 4"x4"
When the spacing between coils is only 3" use 3/4" square or round iron bars as long as the coils. These bars are placed on the coil straps
Angle Iron
B
A
B
2"x4" or 4"x4"
Hole for bolt
2"x4"
When the spacing of the coils is greater than 3" angle iron spreaders are used, as shown. To determine the height of the angle iron "B" for any certain coil spacing "A" consult the table.
NOTE: Do not put the angle irons to close to the return bends, or it will cause a strain on the coils and might make them lay crooked.

| Dimensions | |
|---|---|
| A | B |
| 4" | 1 3/4" |
| 4 1/2" | 2 1/4" |
| 5" | 2 3/4" |
| 6" | 3 3/4" |

Fig. 197.—Details of Pipe Coil Supports.

are evenly tinned over with the use of a soldering acid made of muriatic acid cut with zinc. The fitting is made up while both are hot, and the recess at the back of the fitting filled with solder. In the larger pipe sizes the solder in this recess should be caulked.

The lithage joint is made up by applying a thin layer of the mixture to the threads of both the pipe and the fittings. The mixture should be made in small amounts as needed in the proportion of $1\frac{1}{4}$ parts of lithage to 1 part of glycerine by volume, and it should be mixed thoroughly. Care should be taken not to have excessive lithage lest the effective area of the pipe be decreased, especially in the case of small pipe.

Much future trouble may be eliminated by care in initial cleaning of the inside of any pipe to be erected so as to get rid of any loose mill scale or loose pipe threads, and it is advised that at the very least the pipe be examined on the inside, and that it be rapped with a light ham-

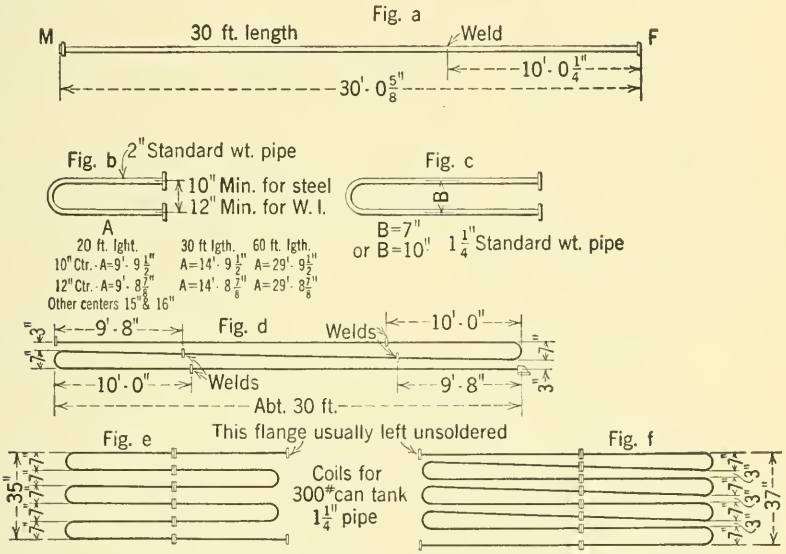



Fig. 198.—Details of Ammonia Coils.

mer when held vertically. Bent pipe should be tested for obstructions by rolling a steel ball through the pipe.

**Piping Details.**—Figures 234 to 239 in Chapter XII give some details of piping, particularly for large installations. In addition, Figs. 197 to 201 give details more applicable to small plants. Piping practice is not standardized, and is given here only as a suggestion. The principal considerations are proper support, and slope of the pipe lines so that drainage of the water pipes and prevention of liquid seal in the ammonia pipes will be provided for, and the proper clearance allowed for so that erection or repairs will be as simple as possible.

The pipe should be very carefully erected, remembering that cold storage piping has a heavy accumulation of frost in most cases, and

that it may also be filled with liquid ammonia. Low points, where liquid may trap in the piping, should be eliminated as far as possible, unless

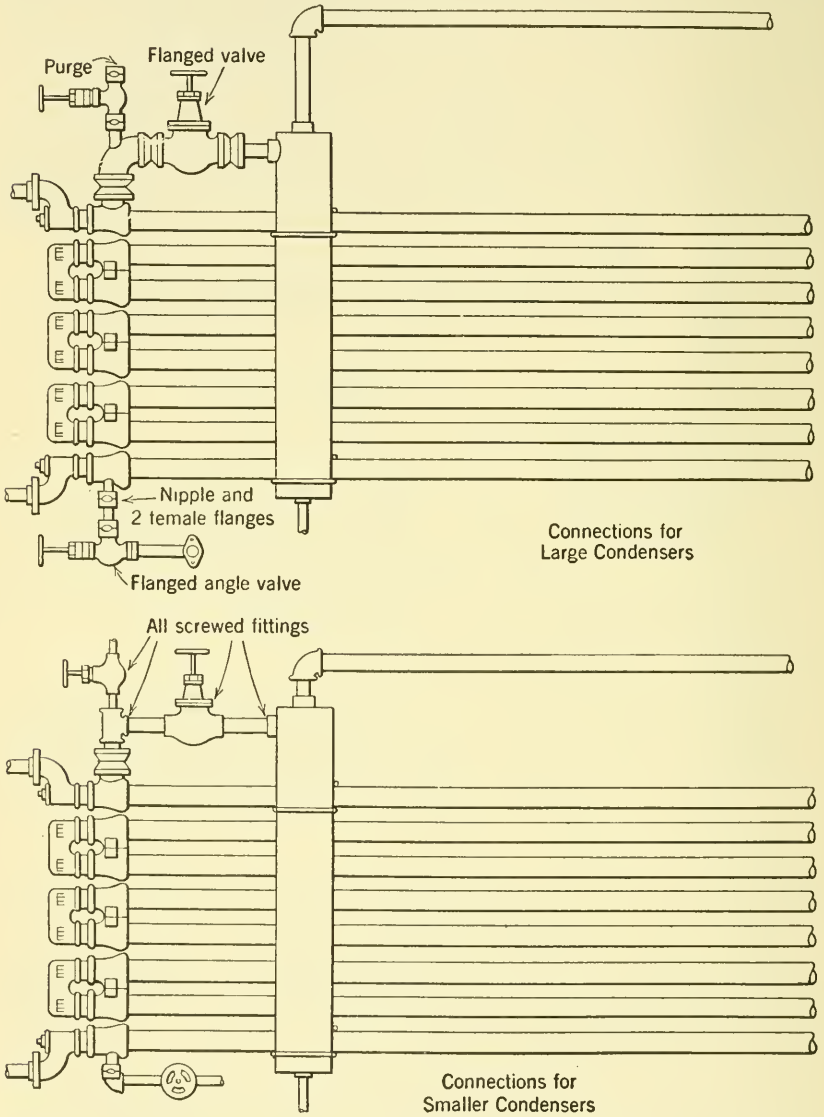



Fig. 199.—Ammonia Condenser Connections.

connection is made at every low point to suitable traps and to the regenerator. Pipe vibration must be kept at a minimum or leaks may

occur at any time. Flanged joints are made up with either lead or oil proof rubber gaskets, and it is wise to chalk-mark the flanges as the

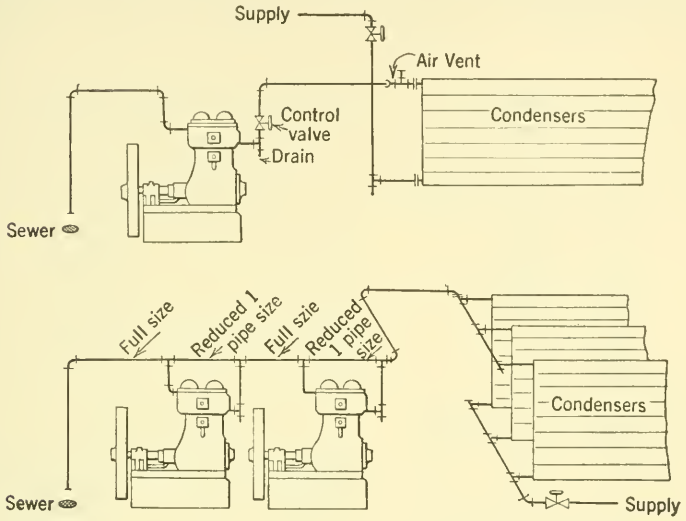



Fig. 200.—Cooling Water Connections.

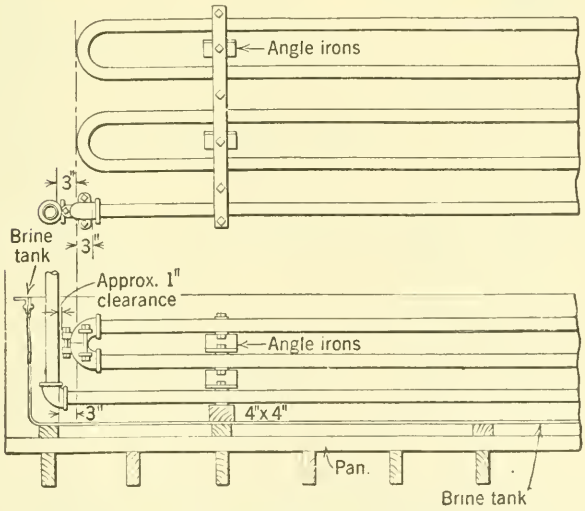
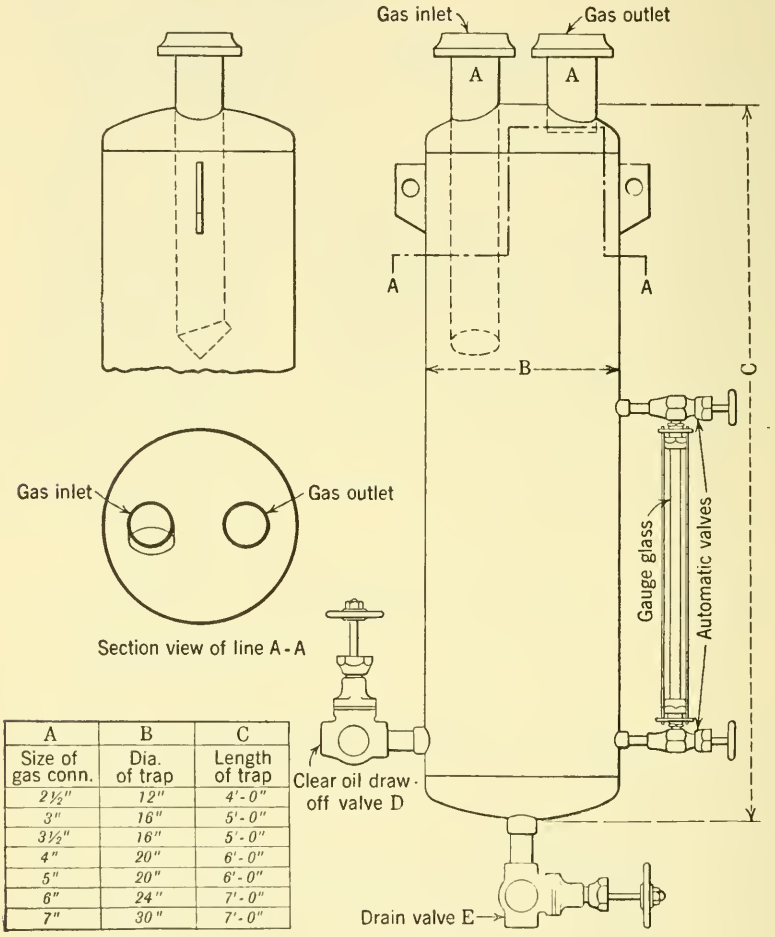



Fig. 201.—Ammonia Coils in Brine Tank.

gaskets are put in place, as a check. The difficulty with lead gaskets is that lead has no elasticity and that it will not give good service

when exposed to the hot discharge gases from the compressor. For the ammonia discharge lines it is better to make use of a high-grade asbestos sheet gasket material which takes without trouble the highest temperatures found in refrigeration and has some elasticity. In making




| A | B | C |
|---|---|---|
| Size of gas conn. | Dia. of trap | Length of trap |
| 2½" | 12" | 4'-0" |
| 3" | 16" | 5'-0" |
| 3½" | 16" | 5'-0" |
| 4" | 20" | 6'-0" |
| 5" | 20" | 6'-0" |
| 6" | 24" | 7'-0" |
| 7" | 30" | 7'-0" |

FIG. 202.—Oil Separator.

up, the bolts must be pulled up evenly all round, and special care must be taken to see that the flange surfaces are free from dirt and scale.

The *oil separator* (Fig. 202) can separate the oil only when it is in a liquid condition. In consequence the separator should be placed as near the condenser as practicable, in order that the gas passing through

may become as cool as conditions will permit. At times it may be wise to install a precooler just ahead of the separator. The precooler is sometimes used in ice-making plants for the purpose of heating water for the drip tank.

The *scale trap* (Fig. 203) is placed in the line especially to protect the compressor. Even with the best of care in erection, mill scale, sand and metal cuttings from the dies will circulate with the ammonia or other refrigerant. The scale trap should be as close to the compressor as possible, and it should be of the size and the design required so that it will function well. At first the need of such a device is very great, but this need decreases as the time elapses. Means should be provided for easy cleaning of the trap with as little air as possible entering the system each time the trap is cleaned.

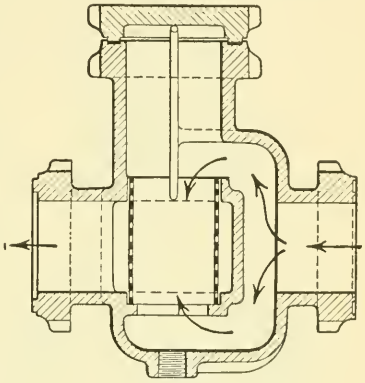


Fig. 203.—Scale Trap.

As soon as the piping has been erected, water and steam or electric connections made, the plant is ready to be tested out for leaks, using air under pressure. The compressor is used for pumping this air pressure, care being taken not to pump up too rapidly at any time for fear of explosions.

Make sure that the compressor is in good shape as regards rod packing, valve action, tightness of bearings, etc. See that the lubrication is in good shape and that the proper kind of oil is used in the cylinders.

**Cylinder Oil.**—It is very important that the proper ice machine oil is used for the cylinder and the crankcase in the enclosed type of compressor. As a rule a freeze test of −20 deg. F. is sufficient, but in some cases—the carbon dioxide compressor when working on low temperatures, for example—it is worth while to get a lower freezing test temperature for the oil, say, −30 deg. F. The splash type of enclosed machine should be filled with oil up to the mark on the frame of the compressor.

**Starting Up.**—All refrigerating machines are designed so that the compressor may be used to pump out the system normally under condenser pressure and to discharge this gas into the low pressure side. This is accomplished by means of by-pass valves cross connected (Figs. 204 and 205). Most compressors are also fitted with a by-pass from the discharge into the suction, especially motor driven compressors, in order

that the required starting torque shall be as small as possible. Each cylinder also has a stop valve for the suction and the discharge line close up to the cylinder. There is required also an opening (a valve or a plug) to the atmosphere on the suction and the discharge side of the cylinder. All of these connections are required at different times during the operation of the compressor.

**Pumping an Air Pressure.**—Open the connection to the atmosphere on the suction side and close the suction stop valve. To put pressure on the high side only, keep the expansion valve closed; to test out the entire system, have the expansion valves open, and to test out the low

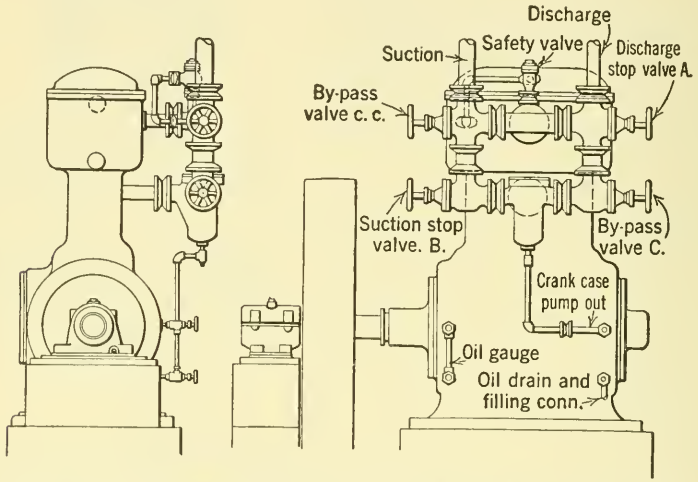



Fig. 204.—By-Pass Valves for the Vertical Single Acting Compressor.

pressure side only, have the cross connections open—with the discharge open to the atmosphere—and the expansion valves closed.

With water passing through the jackets operate the compressor slowly (or intermittently) and raise the pressure on the system to about 150 lb., unless large leaks evidenced by the noise of escaping air appear before this pressure is reached. In a new plant the first things to look for are split pipe, flanges with defective or even without any gaskets, leaky valve bonnets, valve stems, etc. At first the escape of air can be heard, but when that is no longer true painting with soap and water must serve. The latter is only good for small leaks—all others can be heard if there are no other noises in the compressor room. The lather should be put on the pipe and fittings freely, and the pressure raised to 200 lb. at this point, for the entire system, safeguarding the low pressure gage by shutting it off.

When all leaks have been located and repaired the next step is to make a duration test by pumping up to 200 lb. and allowing the pressure to remain on for 12 to 24 hours. Any drop of pressure not accounted for by a lowering of the temperature of the air is due to leaks and should not be allowed to be appreciable.

**Blowing Out the Coils.**—The piping is usually designed so as to permit the *blowing down* of each set of coils to permit freeing the pipes of any loose dirt, steel chips, etc., which may be in the line. For this purpose use pressures of 250 to 275 lb. and open freely to the atmosphere so that a high air velocity will be acquired. Continue the process until the whole system has been blown down. After blowing down it is well to go over the compressor thoroughly in order to clean out any grit or dirt which has worked into the cylinder (the crankcase of the enclosed machine) and the valves and ports. It is valuable to repeat this process about a week after charging and starting operation.

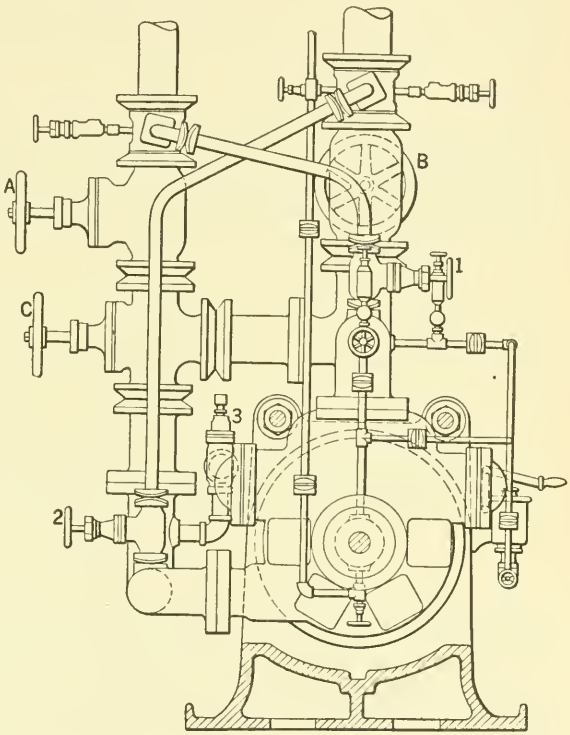



FIG. 205.—By-Pass Valves for Horizontal Double Acting Compressor.

**Test with Ammonia.**—Pump a vacuum on the entire system. This can be done usually by closing the cylinder discharge stop valves (and the by-pass valves), opening all lines and expansion valves and opening the discharge to the atmosphere by means of the valve or plug arranged for that purpose. Pump as good a vacuum as possible, as it is easier, and cheaper, to free the pipe system of air at this point than after charging.

Attach the ammonia charging drum to the charging connection and permit liquid to flow slowly into both the low pressure and the high pressure side. Permit the pressure to rise to about 50 lb. and then look for leaks, first by the smell and the noise of the escape, and then by means of the sulphur stick. When the sulphur dioxide gas formed by the burning of the sulphur stick comes in contact with ammonia gas there is formed a *white* smoke, the indication of an ammonia leak. These sticks may be in the form of candles with a small wick in the center, or they may be sulphur-coated heavy twine—the latter prepared by dipping the twine in liquid sulphur. Even pieces of soft wood may be dipped into the liquid sulphur to form a convenient "candle." Small leaks around welded joints can be stopped by caulking and with care this can be done also to malleable fittings. Bad leaks may require the pumping down and the replacing of the defective part. If the bad leaks are only in one part of the system it may be easier to pump down that

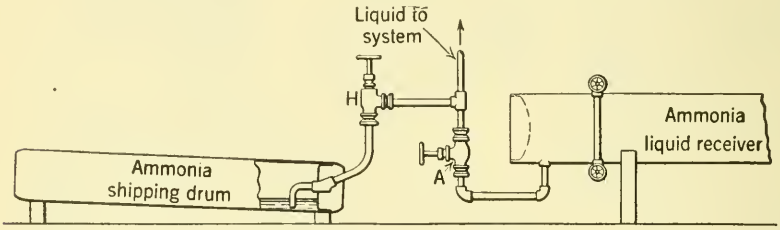



Fig. 206.—Charging Connections.

part only and, storing the ammonia in the other part of the system, to break into the line that needs repairing. The larger plants have separate pump-out compressors and pump-out lines to the different coils or sets of coils, so that repairs can be made without interfering with the operation of the other parts of the plant. The smaller installations, however, need to use the compressor, and the cross connection by-pass valves, to pump out from one part and to store the ammonia in the other part of the system during which operation the plant must be out of commission.

**To Charge the Plant.**—The charge of ammonia should go into the low pressure part of the plant always. Figure 206 shows the proper manner of connecting the shipping drum. To charge, the expansion valve on the line should be opened wide and the valve on the drum should act for the period of charging as the expansion valve. If the evaporator is a brine cooler, the brine *must* be kept in circulation and the compressor and the condenser must be in operation. The first indication of the

emptying of the shipping drum is the appearance of frost on the drum and on the connections to the expansion valve, but the only way to be positive that the drum is empty is to weigh the drum and to compare this weight with the tare on the shipping tag. A sufficient number of drums of ammonia should be charged so that the plant will have enough refrigerant to operate.

**The Amount of Ammonia Charge.**—It is difficult to give exact rules for charging because of the variation in the amount of ammonia used in different cases. One rule,[2] which has been successful in practice, is as follows:

2-in. flooded atmospheric condenser, 12 pipes high, 20 ft. long, 140 lb. per stand.
2-in. standard atmospheric condenser, 24 pipes high, 20 ft. long, 60 lb. per stand.
$1\frac{1}{4}$ by 2-in. double pipe condenser, 8 pipes high, 20 ft. long, 20 lb. per stand.
$1\frac{1}{4}$ by 2-in. double pipe condenser, 10 pipes high, 20 ft. long, 25 lb. per stand.
$1\frac{1}{4}$ by 2-in. double pipe condenser, 12 pipes high, 20 ft. long, 30 lb. per stand.

The ammonia charge required for pipes submerged in brine or water is greater than that required for direct expansion in cold storage rooms. At $22\frac{1}{2}$ lb. per sq. in. suction pressure the following is recommended:

$1\frac{1}{4}$-in. pipe submerged, as in water coolers, brine coolers in ice tanks, $\frac{1}{7}$ lb. per lineal foot of piping.
$1\frac{1}{4}$-in. pipe exposed to the air, $\frac{1}{10}$ lb. of ammonia per lineal foot of pipe.
$1\frac{1}{2}$-in. pipe exposed to the air, $\frac{1}{6}$ lb. of ammonia per lineal foot of pipe.
2-in. pipe exposed to the air, $\frac{1}{3}$ lb. of ammonia per lineal foot of pipe.

It is also usual to figure on the receiver being one-half full.

If one knew definitely the quality of the refrigerant in each part of the system, the calculation of the amount of charge would not be difficult to get accurately for a certain operating set of conditions. The piping from the compressor to the condenser is operated usually with 100 deg. F. of superheat, or more, and from the brine cooler or the direct expansion piping to the compressor it is dry, saturated or superheated a few degrees. The evaporators are always flooded more or less, as is likewise the condenser, but all condensers (except the so-called flooded condenser) attempt to free the piping of the liquid condensate as soon as possible. The vertical and the horizontal shell and tube condenser is a type that frees itself quickly of the liquid. It must be remembered that the charge must be sufficient always to give a liquid seal on the outlet to the receiver, otherwise gas will pass through the expansion valve to the low pressure side of the system thereby choking the coils and reducing the capacity of the plant. There will also result a loss of ice on the return suction line, overheating of the piston rod, etc.

[2] J. Laichinger, National Association of Practical Refrigerating Engineers, 1923.

Usually the operator can also detect by the whistling flow through the expansion valve that gas is passing through, but the better method is to install a gage glass on the liquid receiver connected by means of automatic ball stop valves with the glass tubing. The operator then has an opportunity of observing at all times the amount of liquid seal on the outlet from the receiver. Best results, as regards the capacity of the compressor and of the evaporator can be obtained by frosting back to the compressor when operated at 25 or 30 lb. suction pressure or lower.

Other factors tending to affect the amount of charge required in the refrigerating plant are the pressures carried on the low pressure side and the rate of boiling of the refrigerant in the evaporator. The suction pressure affects the specific volume of the gas so that the higher the pressure the more weight is represented by each cubic foot of volume. Also, in operating, the nearer the temperature of the brine or the air in the direct expansion cold storage rooms is to the boiling temperature of the ammonia the greater the amount of flood it is possible to carry. In extreme conditions it will be found that sometimes "lost" ammonia is found in the coils where no evaporation is taking place on account of the room temperature being too low for evaporation to take place until the liquid *slops* over into the suction return line.

**Operating the Expansion Valve.**—The function of the expansion valve is to feed the evaporating surfaces with liquid. If the load is constant and the compressor is operating at a constant speed, conditions will automatically adjust themselves after a time with any setting of this valve, as the suction pressure will raise or lower until the weight being drawn into the compressor and condensed in the condenser is equal to the amount passing through the expansion valve.

However, each plant has its own correct operating pressure dependent on the duty to be performed. This is such an operating pressure as will give 3 to 5, 10 or even 15 deg. F. difference in temperature between the evaporating temperature of the refrigerant and the temperature of the fluid surrounding the surfaces. The low pressure piping is laid out with the idea in view of having this particular difference of temperature, so that each plant—to do its particular duty—must be operated for the conditions for which it was planned. Incidentally, it is generally understood that a liberal amount of evaporating surface, and a relatively high suction pressure will give best economy to the plant. With a certain amount of piping (if it is clean and operated in the correct manner), if the load cannot be carried, the only alternative is to *decrease* the suction pressure, but this lowers the capacity of the compressor and increases the horse power per ton of refrigeration and therefore such action should be considered the last resort.

**Operation: Removal of Non-condensible Gases.**—In starting up, even with a good vacuum previous to charging, there will be air in the system, and the operation of the compressor will result in bringing this air finally to the condenser where it will remain. If care is taken not to pump a vacuum on any part there will be no occasion for the amount of air or other gases to increase.[3] If care is taken not to pump a vacuum on any part of the plant there will be no occasion for the amount of air or other non-condensible gases to increase in amount. Ammonia or other refrigerants do not disintegrate under usual operating conditions, although some cheap lubricating oils will do so if the temperature of the discharge gases is excessive.

The action of the air in the condenser is to increase the pressure indicated by the condenser gage. This gage registers the total pressure, which is that exerted by the ammonia plus that due to the air present. The pressure of the air is that amount which it would itself exert if occupying the condenser volume by *itself* and at the same temperature. The pressure of the ammonia is that *surface tension* exerted by the ammonia during liquefaction and (if pure) is the pressure corresponding to the temperature of condensation. The law of partial pressures[4] is then

$$p_{\text{total pressure}} = p_{1\text{of the air}} + p_{2\text{of the ammonia}}$$

The "stuff" in the condenser is a mixture of a gas and a vapor, in different percentages in different parts of the condenser, but still a mixture. If air is present the film surrounding the condenser surfaces is nearly pure air while the condenser is in operation. The densest air is where the condensing water is coldest as a rule, but the non-condensible gases are naturally swept along in the direction of the flow of the ammonia. Purging, to free the condenser of foreign gases, should be at the point of greatest air density. This would be the upper part of the shell and tube condenser, and the bleeder condenser, but it is the lower part of the double pipe and the common atmospheric condenser.

Purging should be given considerable care. Without question a large amount of unnecessary purging is done and under no circumstances,

[3] Even sharp freezers requiring −20 deg. F. can secure this temperature without having a vacuum on the low pressure side of the compressor. The reason for air leakage into the compressor when operating under a vacuum is that the stuffing box on the rod will usually leak air *into* the cylinder. Pumping a vacuum on shutting down is very bad practice. Modern machines, even enclosed compressors, can take 150 lb. on the low pressure side without difficulty. It is advised on shutting down that the evaporator coils be pumped down to not lower than 5 lb. gage.

[4] See Chapter III.

whether a purging device is used or not, is it possible in *practice* to purge without losing ammonia. The easiest way to purge and yet conserve the ammonia is to " freeze " the ammonia out of the mixture by the use of refrigerating coils, made up either as a shell and coil or as a double pipe arrangement. All that is required is to cool the mixture to as low a temperature as practical, under which conditions the partial pressure of the ammonia will be reduced and the *weight of the ammonia per cubic foot* of mixture will be reduced in proportion to the density of the ammonia at that temperature. This method will not free the gas being purged out of the system of all the ammonia, and hence it is best to free the system of air and not permit it to get in by careless operation or by the unnecessary opening of the ammonia pipe lines for repairs. At the present time there are several patented devices to assist in purging, although the purge line extending into a bucket of water is the best arrangement for the small plant. Figure 207 shows the Hill non-condensible gas separator which is designed for automatic operation with a relief valve set at 100 lb. at the top of the shell.

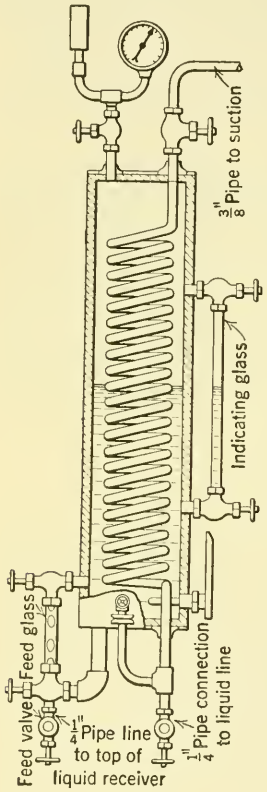

3/8" Pipe to suction
Indicating glass
Feed glass
Feed valve
1/4" Pipe line to top of liquid receiver
1/4" Pipe connection to liquid line

Fig. 207.—Purge Drum and Connections.

**The Regenerator.**—It is undoubtedly true that after the plant has been in operation for some time a certain amount of lubricant passes around the cycle continually. The oil separator does not separate all of the oil and the liquid in the liquid receiver does not separate like oil and water, but forms (to some extent) a solution of liquid ammonia and oil. The result is that oil will get into the low pressure piping, as well as water from the oil, from the ammonia, and from a number of other sources.

In order to get rid of the water and the oil the large plant uses a regenerator (Fig. 208) which operates on the principle of boiling out the ammonia at suction pressure. Warm water is usually used to keep the regenerator warm and connections are made to this device from all low points in the piping. The action of bleeding off should be done at frequent intervals; the ammonia returning at suction pressure and some-

what superheated back to the compressor. The residue consists of water, oil, solid materials and some ammonia.

**Defrosting the Low Pressure Piping.**—It is a mistake to believe that frost on the piping will give—on account of the larger diameter—a greater heat transfer than would be possible with bare pipe. Frost will decrease the heat transfer until finally the evaporator surfaces will fail to give results.

There are a number of ways of defrosting pipes:[5] The air in the room may be permitted to rise in temperature, or the refrigerant may be

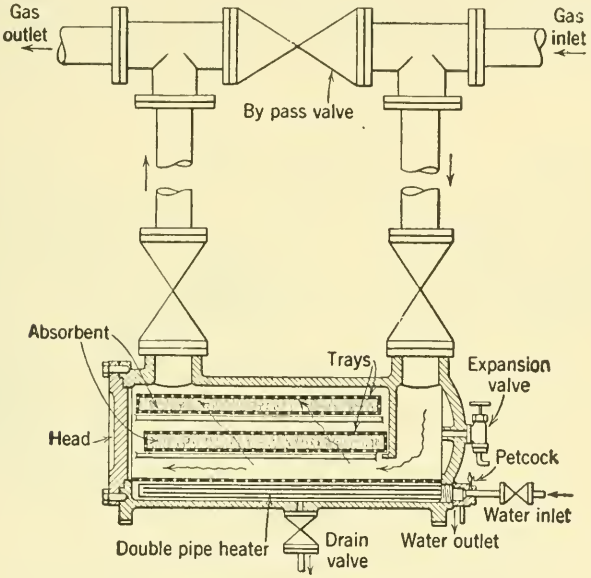



FIG. 208.—The Regenerator.

shut off or the machine stopped for a short time, the snow may be brushed off or the ice chipped off or defrosting may be accomplished by the use of hot gases from the compressor. In the last mentioned, the preferred method in some plants, the frosted pipes act as a condenser, and by the use of the by-pass valves, or by the use of a special compressor, the hot, superheated gas is pumped into the coils to be defrosted (Fig. 209). The larger plants make use of special defrosting lines which can be used at other times for the purpose of pumping out. Another advantage of the defrosting pipe line is that the warming of the piping permits the congealed oily mixture of mill scale, oxide, sand, etc., with the lubricator oil to flow to the traps at the low points of the

[5] See Chapter VIII.

piping. If traps are not installed, and it is desired to clean out the expansion piping, it is necessary to break into the piping, apply steam and then compressed air and finally blow out the line as at the time of erection. If there is any reason to believe that oil is congealed in parts of the condenser, it can be loosened by partly shutting off the water; but the better way would be to open up the two ends and then pass steam followed by compressed air through the piping, as just mentioned.

**Lack of Capacity.**—The refrigerating system is a closed one, using the refrigerant over and over, but the system is subject to troubles inherent in the manner of operation. Scum, dirt and scale may accumulate on the water side of the condenser due to impure water, frost and oil may cover the outside and the inside respectively of the evaporator

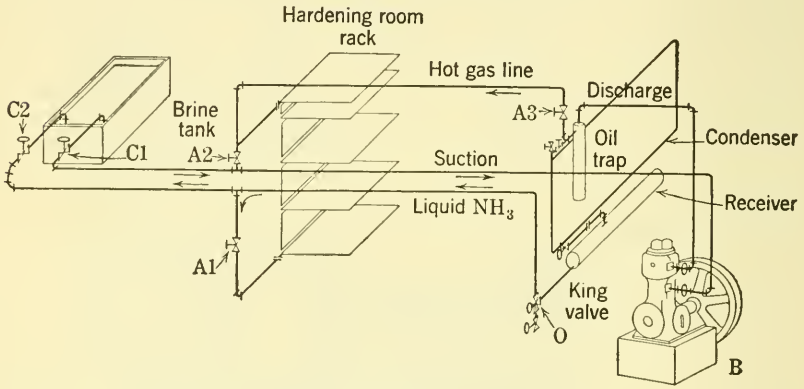



Fig. 209.—Defrosting Piping.

surfaces; leaky valves, piston rings or scored cyclinder walls may result in poor pumping ability in the compressor; obstructed circulation of the water or the ammonia may put the condenser out of commission or at least decrease its working ability; and faulty action in the brine cooler or the direct expansion piping may result in a poor heat transfer. An example of an improvement in operation is as follows:

*a.* Removal of a long accumulation of ice and frost permitted the machine to be operated at 15 lb. suction pressure gage instead of 3 lb. gage with equally good room temperatures.

*b.* Removal of a long accumulation of oil in the pipes permitted a further rise in the suction pressure from 15 to 17 lb. gage.

*c.* The piping had now been returned to normal conditions. This particular job was under-piped, and by an increase of the piping the proper room temperature was obtained by operating at 22 lb. gage.

Referring to curves showing the capacity of ammonia compressors under various operating conditions it will be seen in this case that a compressor (under 150 lb. condenser pressure) will require 9.2 cu. ft. per minute piston displacement per ton of refrigeration at 3 lb. gage and 3.7 cu. ft. at 22 lb. gage. With these two conditions the same compressor would give nearly 2.5 times the capacity at 22 lb. as at 3 lb. suction pressure.

**Withdrawing the Charge.**—Small and even moderate-sized plants are supposed to have receiver capacity, with what can be stored in the condenser or the evaporator coils, to store all of the ammonia charge in order that repairs can be made when needed. The larger plants cannot

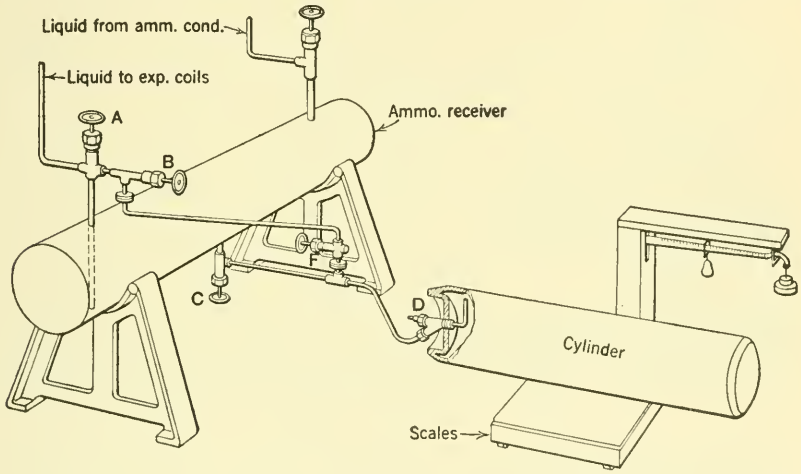



Fig. 210.—Withdrawing the Ammonia Charge.

do this, and withdrawal of part of the charge becomes necessary. In order to withdraw the ammonia an arrangement similar to Fig. 210 can be used. Note that in the figure the shipping drum has the bent tube pointing upwards so as to be able to free the drum of the air and the gas after it is partly filled with liquid. The method is to pump down the pressure by closing the valves *A* and *C* and opening *B*, *F*, and *D*. After the pressure in the drum has been lowered, close *F* and open *C*, repeating the process until the net weight in the drum is the amount the drum was designed for, usually 50 or 100 lb.

In withdrawing the ammonia charge from the system it is not necessary to use the complicated apparatus just mentioned. Usually it is sufficient to proceed as follows:

Connect an empty shipping drum to a point between the liquid

receiver and the expansion valve, with the valve on the shipping drum pointing down. Open the connections to the shipping drum, and cool the drum with cold water or ice. Note when the drum is full by weighing frequently. Particular care must be taken not to fill the drum beyond the safe amount indicated by the shipping tag.

**Causes of Excess Pressure.**—Excess pressure in the condenser may be caused by:

*a.* Too small a condenser, hot cooling water, or too small an amount of water for the temperature of the water used.

*b.* Inert gases in the condenser, mostly air or decomposed lubricating oil.

*c.* Poor heat transfer, due to dirt and scale from the cooling water, and oil and dirt on the ammonia side of the condenser.

*d.* Condensers partly filled with liquid ammonia due to an over charge (for the suction pressure being carried).

*e.* One or more stands not functioning due to broken or faulty valves so that the liquid cannot drain properly.

**Causes of Loss of Capacity.**—Loss of capacity in the plant may be caused by:

1. Insufficient piping surface, or frost on the outside and oil and scale on the inside.

2. Leaky valves, or safety head, piston rings or scored cylinders in the compressor; slow closing of the valves due to a number of causes.

3. Obstructions in the suction lines or broken valves preventing full opening.

4. Too small a charge of ammonia, thereby permitting gas to pass through the expansion valve.

5. The presence of water in the evaporating coils, the action of which is to require a lower pressure to be carried in order to obtain the temperature expected of anhydrous ammonia.

Figures 212 and 213 may be used as a convenient check on the capacity and other characteristics of small twin cylinder, single-acting vertical ammonia compressors for the rotative speeds indicated, which are nominal. For other speeds the results will be proportional. Tables 81 and 82 give details of valves, water and ammonia connections, etc., for the York enclosed compressor. These last details are very nearly standardized and are approximately true of all American manufacturers.

**Starting Up.**—The usual procedure in starting up the compressor is as follows:

1. Turn the water on the condenser and the compressor water jackets.

2. Start the brine pump, and (if a centrifugal pump) make sure that the brine is flowing.

3. See that the lubricators are in shape, and give the ammonia cylinder a small amount of cylinder oil by hand.

4. See that the suction and the discharge valves on the machine are closed, and open the by-pass from the discharge to the suction.

5. Start the compressor, and

6. Open the suction and the discharge valves. Close the by-pass valve.

7. Adjust the expansion valve so as to get proper operating conditions.

**Testing for Leaks.**—Ammonia losses[6] are greatest through the piston rod packing, valve stems, and flanged joints. It is next to impossible to keep the piston rod packing (Fig. 211) tight if the rod is not centrally located at all times in the stroke. Carelessness in taking up wear in the crosshead shoe is responsible for trouble of this sort. Also, the piston rod will wear in time more in the middle of the stroke than at the ends. Wear in excess of $\frac{1}{64}$ in. should be given attention, and probably will require turning down. It is wise to keep all valve stems well lubricated, and occasionally tighten on the packing. The sulphur stick is the best test for such leaks, a leak being indicated by white fumes, as already stated.

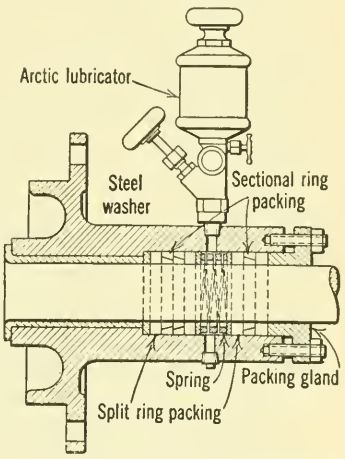


FIG. 211.—Piston Rod Packing.

The shell and tube and the atmospheric condenser can be tested with the sulphur stick while the outside is dry, and the pressure is up to about 200 lb. The double pipe condenser can be tested in the same manner, but the usual manner is to test the cooling water with Nessler's solution, which turns yellow (or, for heavy leaks, deep brown) when ammonia is present in the water. The same test is equally good for the brine cooler or the brine (ice) tank as well as the jacket water of the compressor. Nessler's solution can be made by dissolving 17 grains of mercuric chloride ($HgCl_2$) in 300 c.c. of distilled water and 35 grams of potassium iodide (KI) in 100 c.c. of distilled water. Add these, stir until a slight red precipitate is formed, and then add a cool solution of 120 grams of

[6] Average ammonia losses, according to H. T. Whyte, of the Consumers Ice Co., Chicago, cost 2.3c. per ton of ice, and Charles Neeson of the Knickerbocker Ice Co. New York, states that the average of 14 plants in Philadelphia is 2.0c. per ton.

potassium hydrate (KOH) in 200 c.c. of water and then dilute with water until 1000 c.c. is obtained. Add this to the $HgCl_2$ solution until

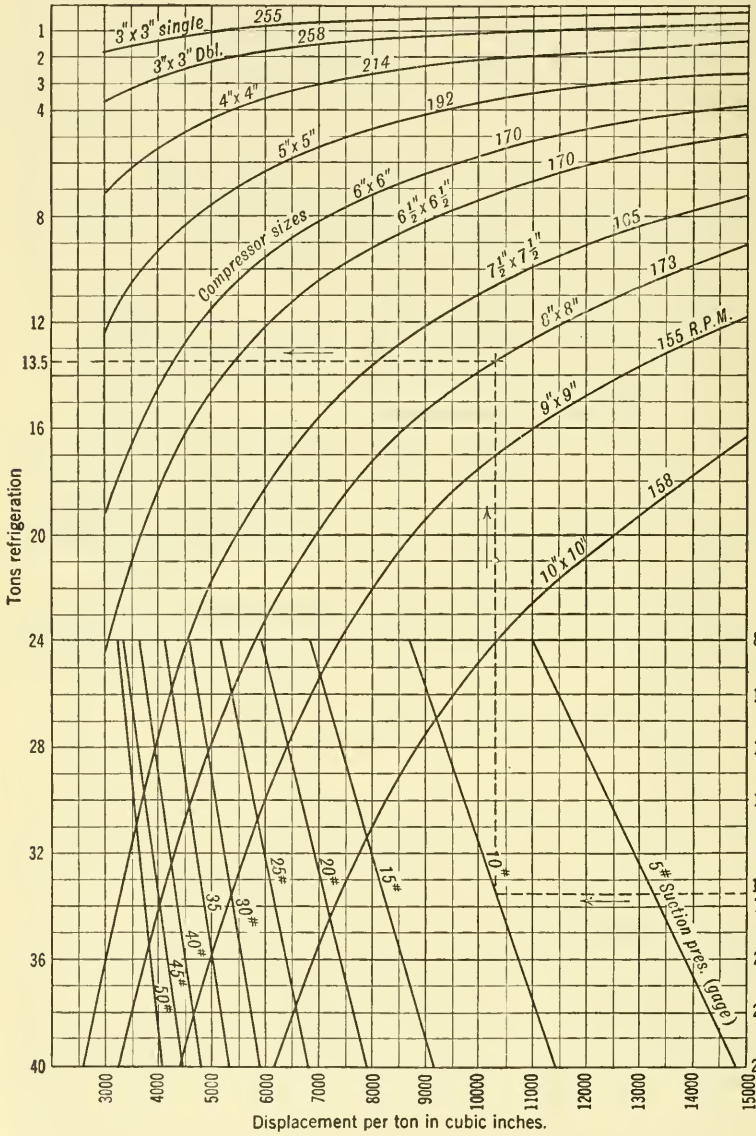



FIG. 212.—Capacity Curves—Enclosed Type Ammonia Compressors.

a precipitate is formed and then let it stand until it is settled and finally pour off the clear liquid into a blue bottle in order to prevent decomposition.

When testing calcium brine for ammonia it is necessary to precipitate the calcium out of solution first by adding to the sample of the brine

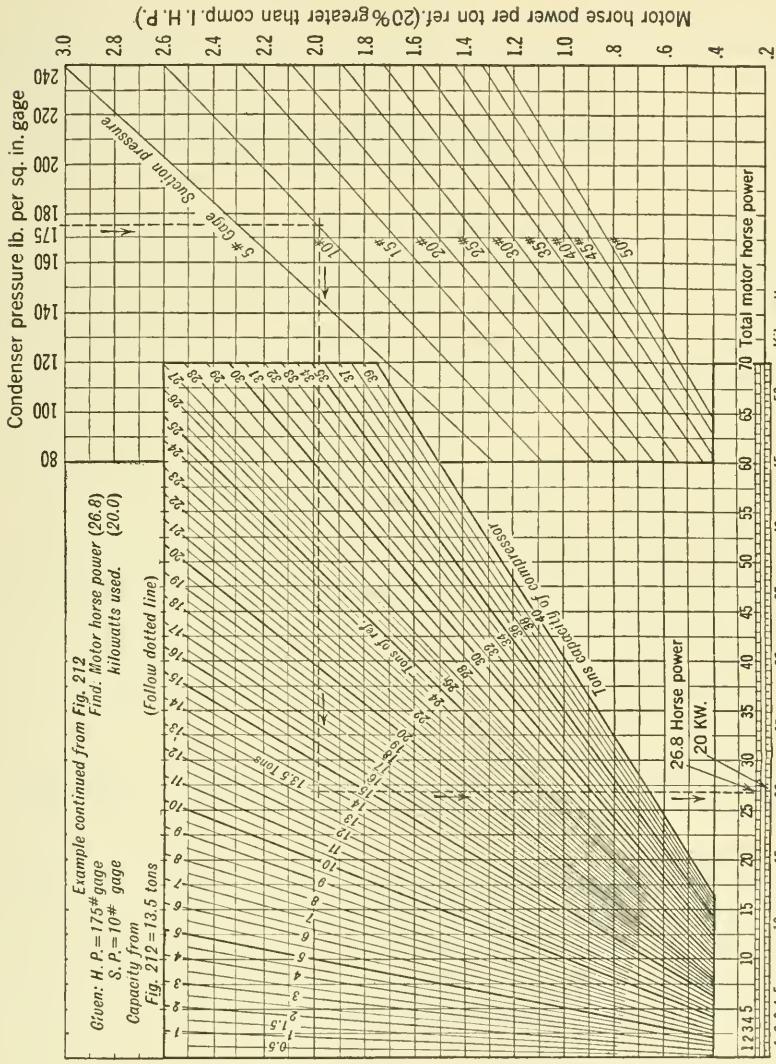



Fig. 213.—Power Requirements—Enclosed Type Ammonia Compressors.

diluted with water enough concentrated solution of sodium carbonate ($Na_2CO_3$) to precipitate out the calcium, and then add to the filtrate Nessler's solution. A change to a brown-colored solution indicates the presence of ammonia in the original solution.

**Carbon Dioxide Systems.**—In carbonic refrigeration the erection is practically the same as for ammonia. In charging the plant it has not been the custom to pump a vacuum as is always done in the case of ammonia, and in consequence there is considerable air in the piping after the operation of charging has been completed. Heretofore it has been considered satisfactory to rely on the numerous leaks in the piping to remove the air but it is quite evident that this custom will need to be abandoned and that provision will have to be made to pump out the air, at least partially, as in the case of systems with nominal condenser pressures.

**Charging.**—The amount of the charge in a carbonic system can be estimated in a manner similar to that in the ammonia plant by a calculation of the internal volume of the condensers, the piping on the high pressure and the low pressure sides and the amount of the liquid which should be in the condenser and the liquid lines. By the selection of the specific volume of the vapor for the pressures expected to be found in the pipes a fair estimate may be made of the required charge. The usual manner, however, has been the rather crude method of estimating the charge by the discharge pressure, which should approximate saturation for the temperature of the outgoing condenser water. If the condenser pressure is considerably lower than this amount then the charge is insufficient and more gas must be added. No gage glass is used on the liquid receiver of the carbonic plant, as such a glass would need to withstand pressures in excess of 1000 lb. per sq. in. nor has it been the custom to install liquid receivers of sufficient size to warrant the name.

The lack of a liquid receiver is surprising inasmuch as a pound of ammonia will produce as much refrigeration as 7 or 8 lb. of liquid carbon dioxide, and therefore the possibility of keeping the expansion valve sealed with liquid is decreased in proportion. For example, suppose that with ammonia it is usual to have a liquid receiver containing 100 lb. of liquid in order to supply the demands of the service imposed on it and still have a complete liquid seal at all times on the expansion valve. For the same variation of load there would be required some 700 lb. of carbon dioxide and it is positive that such a liberal charge is never supplied. Such being the case it is very likely that a considerable loss of capacity is frequent due to a loss of the liquid seal. The absence of a glass gage on the lower part of the condenser or the liquid receiver makes it necessary to rely on the pressures with the result that the careless operator permits the plant to run inefficiently. There is no question that this method of guessing at the proper operating conditions should be removed from carbonic refrigeration. To do this a suitable

liquid receiver should be installed and a liquid level indicator (for example a hollow steel ball within a brass tube, the location of which can be shown by means of a magnetised indicator.) There are also special small bore glass tubings manufactured at the present time which will stand from 1200 to 1400 lb. per sq. in. with safety and these can be used in liquid gage glasses if desired.

The action of *liquid* carbon dioxide in the refrigeration system is practically identical to that of ammonia and other volatile liquids where the liquid boils in the evaporating coils in absorbing heat and thereby becomes a vapor. The only disconcerting factor is the very low critical temperature (88.0 deg. F.) above which the gas cannot be liquefied even with excessive pressures. However this fact does not mean that no refrigeration is possible at condenser pressures greater than that corresponding to the critical temperature, but simply that the refrigerant flowing to the expansion valve is not a liquid although it is a very dense gas. The P–I diagram, shown in Chapter 7, indicates that when operating above the critical temperature a loss in the refrigerating effect per pound of carbon dioxide is inevitable but that otherwise there is no difference between carbonic and ammonia refrigeration.

In marine practice it is usual to overcharge the system during the passage through the tropics. The reason for this is that an overcharge, as the refrigerant does not condense, means a heavier discharge pressure which, for a stated temperature of the refrigerant at the expansion valve, means a smaller value of I at this point in the cycle. With the excess charge more work is performed on the gas during compression but the capacity is increased also. The actual pressure in the condenser under these conditions has no connection with the condensing water and it may be 1200, 1300 or 1400 lb. per sq. in. depending on the amount of the overcharge.

Great care must be exercised to operate with the proper sized opening of the expansion valve. If this pressure reducing valve is open too much the liquid seal is broken and partly cooled gas is permitted to pass into the evaporating coils. The effect of this action is an increased evaporating pressure and a reduced condenser pressure, the capacity of the plant becomes decreased and the desired temperatures in the evaporating coils will not be obtained. Charging more gas into the system will not give any satisfaction.

A more recent and advantageous method of operation, however, is the dual system, or the operation of the compressor by admitting two pressures into the cylinder. This is done by admitting the low pressure gas from the evaporating coils in the usual manner but at or near the end of the suction stroke ports in the cylinder are uncovered with

register with a suitable connection to the high pressure evaporating coils. When the piston overrides these ports in the cylinder the gas under the higher pressure flows in, the low pressure suction valve closes and the gas in the cylinder increases in pressure at constant volume. During the compression stroke the total weight of the gas in the cylinder is compressed, and this is considerably more than would have been the case if the low pressure suction gas only had been compressed. The manner of operation is as follows:

In the dual system a part of the refrigerant from the condenser which may be a liquid at 80 or 85 deg. F. or a dense gas at 90 or 100 deg. F. is passed through suitable cooling coils made cool by a fraction of this same carbon dioxide from the condenser which is regulated to absorb heat at a temperature of approximately 50 or 60 deg. F. The vapor resulting from the cooling of the refrigerant is at a higher pressure than that of the evaporating coils performing useful refrigeration and this gas enters the cylinder at the end of the suction stroke as already mentioned.

The advantage of the dual system is very evident. The refrigerant leaving the condenser may be at a high temperature and yet by the method just described it may be cooled to some temperature between 50 and 60 degrees. The same weight of refrigerant enters the evaporating coils in each case, but the amount of refrigeration per pound of the carbon dioxide is greater with the colder temperature at the expansion valve. The dual system requires more power for compression than the simple system.

**Operating the Absorption Machine.**—Referring to Chapter 3 it will be seen that a definite relationship exists between the temperature, concentration and the pressure of the aqueous solution in the generator. For proper operation at a particular steam pressure in the steam heating coils there must be a definite average concentration of the aqua in the generator. If the solution is too weak then capacity cannot be obtained without raising the steam pressure. Also the proper amount of aqueous solution must be circulated by means of the strong aqua pump (Table 15).

To charge the system a vacuum should be pumped on the entire system as in the compression plant, by using some form of vacuum air pump for the purpose. The refrigerant is charged in the form of aqua until the generator, absorber and exchanger have their proper amounts, then with anhydrous ammonia until the condenser, liquid receiver, evaporating coils and the return line to the absorber have the proper amount of ammonia for operating conditions. A calculation similar to that given for the compression plant can be made for the absorption plant.

**To Operate.**—In operation the following routine should be followed:

*a.* Turn on the water to the condenser, absorber, rectifier and weak aqua cooler.

*b.* Start the strong aqua pump at the rated speed, see that the weak aqua is flowing properly and that the weak aqua regulator is operating.

*c.* Start the brine pump, if there is one. Note the absorber pressure and consider whether it is the amount desired and to be expected with the temperature of water supplied to the absorber.

*d.* Turn the steam on the generator steam coils and adjust the pressure required by the design of the generator and the concentration carried. Note the amount of liquid in the liquid receiver and estimate whether the amount is sufficient.

*e.* Open and adjust the expansion valve.

In shutting down use the reverse order of operations.

Some disintegration occurs in the generator and inert gases are formed. To decrease this action a solution of potassium dichromate using 0.2 lb. per 100 lb. of aqua should be added to the generator.

TABLE 81

SUCTION AND DISCHARGE VALVES AS USED ON VERTICAL MACHINES

| Size of Machine, Inches | Suction Valves | | | Discharge Valves | | |
|---|---|---|---|---|---|---|
| | Number of valves | Diameter of opening, inches | Lift, inches | Number of valves | Diameter of opening, inches | Lift, inches |
| 3 × 3 | 1 | $1\frac{9}{16}$ | $\frac{1}{4}$ | 1 | 3 | $\frac{1}{8}$ |
| 4 × 4 | 1 | $2\frac{5}{16}$ | $\frac{3}{8}$ | 1 | 4 | $\frac{3}{32}$ |
| 5 × 5 | 1 | $3\frac{1}{8}$ | $\frac{3}{8}$ | 1 | 5 | $\frac{1}{8}$ |
| 6 × 6 | 1 | $3\frac{5}{8}$ | $\frac{5}{8}$ | 1 | 6 | $\frac{7}{64}$ |
| $6\frac{1}{2}$ × $6\frac{1}{2}$ | 1 | 4 | $\frac{5}{8}$ | 1 | $6\frac{1}{2}$ | $\frac{7}{64}$ |
| $7\frac{1}{2}$ × $7\frac{1}{2}$ | 1 | 5 | $\frac{5}{8}$ | 1 | $2\frac{3}{4}$ | $\frac{3}{8}$ |
| 8 × 8 | 1 | $5\frac{1}{4}$ | $\frac{1}{2}$ | 1 | $3\frac{1}{2}$ | $\frac{3}{8}$ |
| 9 × 9 | 1 | $6\frac{1}{8}$ | $\frac{1}{2}$ | 1 | $3\frac{1}{2}$ | $\frac{3}{8}$ |
| 10 × 10 | 1 | $6\frac{3}{4}$ | $\frac{1}{2}$ | 3 | $2\frac{1}{4}$ | $\frac{3}{8}$ |

## TABLE 82

### York Y-15 Compressor Data

| Size of Machine | Capacity in Tons | Speed in R.p.m. | Diameter of Shaft, Inches | Connections, Inches | | | | By Pass, Inches | Weight Without Flywheel Pounds | Flywheel | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Ammonia | | Water | | | | Diameter, inches | Face, inches | Weight, pounds |
| | | | | Low | High | In | Out | | | | | |
| 3×3 *S* | ¾ | 255 | 1¾ | ½ | ½ | ½ | ½ | ¼ | 550 | 30 | 4 | 300 |
| 3×3 *D* | 1½ | 258 | 1¾ | ¾ | ¾ | ¾ | ¾ | ¼ | 760 | 30 | 4 | 300 |
| 4×4 *S* | 1½ | 215 | 2 | ¾ | ¾ | ¾ | ¾ | ¼ | 640 | 32 | 4½ | 360 |
| 4×4 *D* | 3 | 214 | 2 | 1 | 1 | ¾ | ¾ | ¼ | 990 | 32 | 4½ | 360 |
| 5×5 *D* | 5 | 192 | 2½ | 1¼ | 1¼ | ¾ | ¾ | ⅜ | 1435 | 36 | 6½ | 600 |
| 6×6 | 8 | 170 | 3 | 1¼ | 1¼ | ¾ | ¾ | ⅜ | 1960 | 48 | 7 | 700 |
| 6½×6½ | 10 | 170 | 3½ | 1¼ | 1¼ | 1 | 1 | ⅜ | 2125 | 48 | 8 | 825 |
| 7½×7½ | 15 | 165 | 4 | 1½ | 1¼ | 1 | 1 | ⅜ | 3000 | 50 | 10 | 1300 |
| 8×8 | 20 | 173 | 4½ | 2 | 1½ | 1 | 1 | ⅜ | 4300 | 60 | 10 | 1400 |
| 9×9 | 25 | 155 | 5½ | 2 | 1½ | 1¼ | 1¼ | ⅜ | 6050 | 70 | 10½ | 1500 |
| 10×10 | 35 | 158 | 6 | 2½ | 2 | 1¼ | 1¼ | ½ | 8800 | 78 | 12½ | 2000 |

## CHAPTER XI

## THE TESTING OF REFRIGERATING PLANTS

The testing of refrigerating plants should not be a very different process, in principle, from that used for any other kind of plant except that the working medium is usually a noxious gas and is not permitted to leave the metal walls of the retaining pipe system or other apparatus. On account of this property of the refrigerant, and because the media employed are all quite expensive, due care must be taken to prevent accidents and undue loss. The testing consists in the measuring of the temperatures and pressures, the weights of volumes of the water, liquid refrigerant, and brine, the indicated horse power of the compressor, the revolutions per minute, and other quantities affecting the capacity of the plant.

In testing refrigerating plants the investigation may be for the entire plant, or it may be confined to a study of the compressor, the condenser, or the low pressure side. A study of the simple compressor would probably involve the action of the valves and ports, the degree of tightness of the piston rings and valves, and possibly the temperature of the discharge gas. In more complex machines, as in the case of dual compression or stage compression, the investigation could very well be amplified to include methods of varying the operating conditions and the economical methods of operation.

The condenser furnishes an excellent field for investigation at the present time. The important points are, as regards improving the effectiveness of the surface supplied, reduction of the amount of water required, and the reduction of the condenser pressure. The problem becomes one of securing values for heat transfer and of measuring the amount of water used and the refrigerant liquefied. At the present time (1927) little has been done on air-cooled condensers or on the automotive fan type of air-cooled radiators using water on the condenser.

The low-pressure side embraces methods of increasing heat transfer. This investigation requires a knowledge of the amount of the refrigerant boiled from the coils per unit of time and a complete knowledge of the temperatures involved. In most of the investigations on refrigerating systems the temperatures involved are the principal observations to be

made, and in the following the methods of securing temperatures will be given considerable prominence.

**Temperature.**—There are three ways that temperatures, such as are required by refrigerating engineers, may be measured: by the ordinary mercury in glass thermometer, by the electric resistance thermometer, and by the thermocouple. The mercury thermometer is the easiest to use for most work, but it is subject to considerable error. There is always a certain time lag in these readings, and they are of value only when the temperatures are reasonably constant. The greatest error is that due to conduction, especially in superheated gas temperatures. The graduations should be etched on the glass tube, and the thermometer should be calibrated for 32 deg. F. (the melting temperature of ice under standard atmospheric pressure) and possibly at 212 deg. F. (the boiling temperature of pure water under normal pressure). There is a tendency for the bulb to change its shape with time, a fact that affects all readings, and there is a possibility that the bore of the tube is not uniform in cross section, which would affect the intermediate readings.

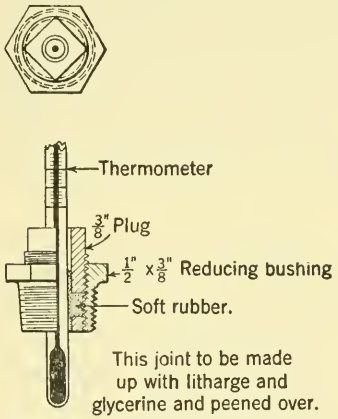

Thermometer
3/8" Plug
1/2" x 3/8" Reducing bushing
Soft rubber.
This joint to be made up with litharge and glycerine and peened over.

Fig. 214.—Thermometer Stuffing Box.

Temperatures requiring accuracy should be secured by means of calibrated thermometers throughout the range required in the test. In refrigeration the temperatures usually desired are between −20 and +40 deg. F. Frequently thermometers can be checked sufficiently well by calibration with a standard thermometer in a bath of oil, immersed up to the point of immersion of the test thermometer under test conditions.[1] The thermometer, for low temperature

[1] If the stem exposure correction must be applied, this may be calculated as follows:

$$\text{Correction} = 0.000085 \times N(T^\circ - t^\circ),$$

where

$N$ = the number of degrees emergent from the bath;
$T$ = the temperature of the bath;
$t$ = the mean temperature of the emergent stem.

*Example.*—If the observed reading is 212 deg. F., the level of the bath (on the thermometer stem) is 100 deg. F. and the mean temperature of the air above the bath is 75 deg. F., then

$$\text{the correction} = 0.000085 \times 112 \times (212 - 75) = 1.3 \text{ deg. F.}$$

work, is best located as shown in Fig. 214, with the stuffing box and the bulb directly in the fluid the temperature of which is desired. The metallic thermometer well (Fig. 215), made of iron for ammonia and brass for other fluids, is a good arrangement for temperatures that do not require great accuracy. When temperatures are taken in this manner, care must be observed to lag the pipe sufficiently on both sides of the well so as to insure against radiation and other losses. It is wise to insert the thermometer in a tee (Fig. 215) so as to eliminate excessive radiation to the walls which may be, under certain conditions, as great as two or more degrees.

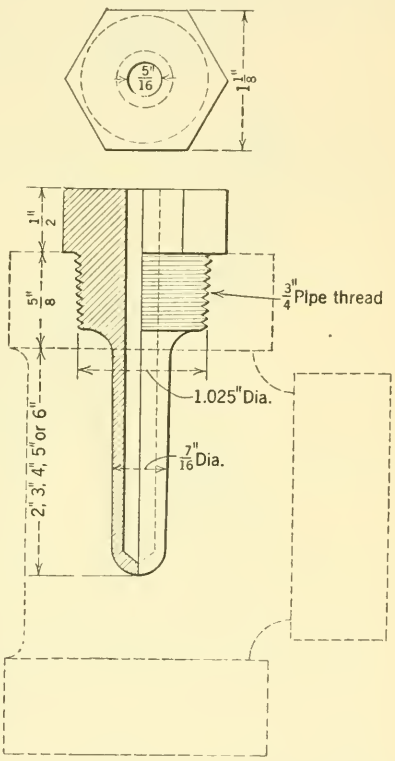



Fig. 215.—Thermometer Well in a Tee.

**The Electrical Resistance Thermometer.**—Resistance thermometers are based on the principle of the change of electrical resistance with the temperature, which means that nothing else—like a strain on the wire or the action of corrosion—should affect the resistance of the wire. For this reason these thermometers use platinum and nickel (the latter for moderate temperatures). In the construction a fine wire is wound on a spool, sometimes about $\frac{3}{8}$ in. diameter, and is sealed in a case.[2] The measurement of the resistance may be done by the use of a Wheatstone Bridge or by the use of a potentiometer, the latter being shown in Fig. 216.

The use of a potentiometer necessitates,[3] "two leads from each of the ends of the resistance, to which they should be soldered or welded. The resistance to be measured is represented by $r_2$, $B$ is the battery to pass a current through $r_2$, $R$ is a standard high-class resistance. The drop of voltage across $R$ measures the amount of current passing through $R$. $r_1$ is the resistance which would be used to bring the current to a suitable value for the size of wire in $r_2$ and the value of $R + r_2$. The

[2] Bureau of Standards, Circulars No. S 407 and No. T 170.

[3] P. Nichols, Temperature Measurements, Refrigerating Engineering, 1924.

potentials, $e$ and $e'$, would be measured on the potentiometer and from Ohm's law,

$$r_2 = \frac{e'}{e} R.$$

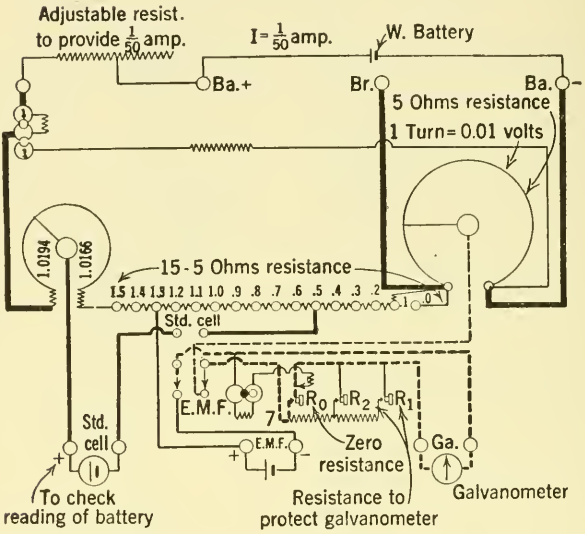



Fig. 216.—Type $k$ Potentiometer.

"There are cases where the resistance principle can be usefully employed, such as where the resistance wire is not in the form of a bulb but is laid on a surface, or imbedded in a material when it is desired to obtain an average temperature over a considerable length or area. For this platinum would usually be too expensive and unnecessary. For short lengths nickel would be used, and for long lengths, copper. To be able to translate resistance values into temperatures, wire with equational relationship between the two would be purchased from an instrument maker, or a piece to be used would be calibrated. If this method is used, an analysis should be made of the readings which are obtained, in order to find the best value for $r_2$, and for the current passed through.

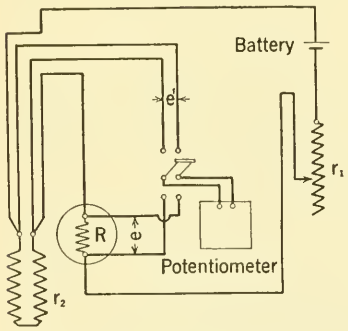



Fig. 217.—Arrangements for Electric Resistance Thermometer.

"The resistance and temperature coefficient—that is, its change of resistance per degree of temperature—of commercial copper is fully covered in B. of Std., S. 147 and S. 148. Commercial soft wire was found to have a fairly constant value in different samples and the temperature coefficient below 212 deg. F. has a linear relationship with temperature, so that

$$R_1 = R_0[1 + a(t_1 - t_0)].$$

where $R_1$ and $R_0$ are the resistances at $t_1$ and $t_0$, and $a$ is a constant for a definite value of $t_0$. The symbol $a$ represents the coefficient for constant mass; that is, we are dealing with a fixed weight of wire and not a fixed length, as the length will change with the temperature. Values for copper are:

| $t_0$ | $a$ |
|---|---|
| 32 deg. F. | 0.00238 |
| 59 deg. F. | 0.00223 |
| 68 deg. F. | 0.00219 |
| 77 deg. F. | 0.00214 |

The value for $a$ within the range of 32 to 77 deg. F. also can be expressed as

$$a = 0.00255 - 0.0000053t.$$

"If, for example, the resistance of a piece of copper wire were measured at 59 degrees and found to be $R$, then the temperature at any other resistance, $R_t$, is found from the equation

$$t_1 = \frac{R_1 - R_0}{0.00223R_0} - t_0.$$

These data are useful under some circumstances, or for changes of temperature, as they only involve measuring the resistances of a piece of wire used at one known temperature. For accuracy, the value of $R_0$ and $a$ should be obtained by finding the resistance of the actual pieces of wire to be used—preferably with the leads attached—at two temperatures.

"Nickel wire is more likely to vary, on account of impurities, and must be calibrated. The equation $\log R = a + bt$ holds approximately, and values for $a$ and $b$ can be determined by tests at two temperatures. For a good commercial wire $a = 0.0119$ and $b = 0.000372$ for $t$ in degrees F. Table 83 will guide in fixing the size of the wire to be used. There is no advantage in too high a resistance, since although the difference in resistance will be larger, yet the sensitivity of the measuring instrument decreases. If the wire is uninsulated the apparent resistance will

be affected by moisture collecting on the wire. For accurate work all wire used should be calibrated."

TABLE 83

Resistance per Foot of Wire at 70 Deg. F.

| Size of B. & S. Gage | Copper | Nickel | Size of B. & S. Gage | Copper | Nickel |
|---|---|---|---|---|---|
| 21 | 0.013 | 0.079 | 31 | 0.130 | 0.811 |
| 22 | 0.016 | 0.100 | 32 | 0.164 | 1.004 |
| 23 | 0.020 | 0.125 | 33 | 0.207 | 1.275 |
| 24 | 0.025 | 0.159 | 34 | 0.260 | 1.619 |
| 25 | 0.032 | 0.200 | 35 | 0.328 | 2.047 |
| 26 | 0.041 | 0.254 | 36 | 0.414 | 2.572 |
| 27 | 0.051 | 0.314 | 37 | 0.522 | 3.183 |
| 28 | 0.065 | 0.405 | 38 | 0.658 | 4.019 |
| 29 | 0.082 | 0.503 | 39 | 0.830 | 5.270 |
| 30 | 0.103 | 0.643 | 40 | 1.047 | 7.144 |

**Thermocouples.**—The principle of the thermocouple is that of the production of a small electromotive force due to the contact of two

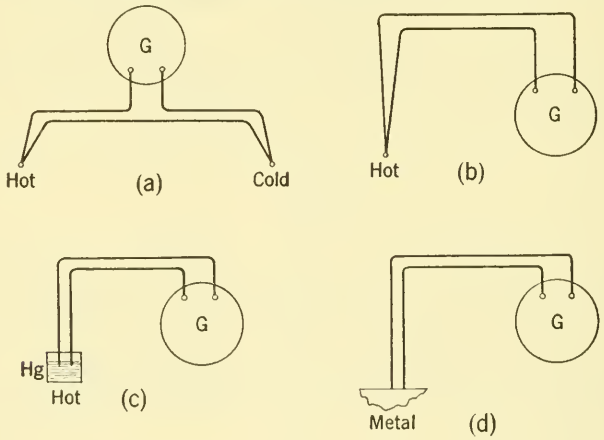



Fig. 218.—Simple Thermocouple Connections.

dissimilar metals which under certain well-defined conditions is a function of the difference of the temperature of the two junctions and of the metals used. A very simple thermocouple arrangement is illustrated in Fig. 218, in which a galvanometer is shown in the circuit, and in cases *b*, *c* and *d* the galvanometer is also the cold junction. In *a* the cold junction may be the temperature of the air in the room or it may be a

container with a mixture of ice and water. The wires must be insulated between junctions. Thermocouples may be used in series (Fig. 220), and the total e.m.f. will be the sum of each separate e.m.f., in which case the usual practice is to divide by the number of the couples in order to find the average, or it can be in parallel (Fig. 220). If a number of couples are in parallel, the resulting reading of the potentiometer will

TABLE 84

COPPER—CONSTANTAN THERMOCOUPLE

E.m.f.—Temperature, With Cold Junction in Ice
Deophysical Laboratory (L. A. Adams)

| E.m.f. Millivolts | Temperature, Degrees F. | Temperature Difference | E.m.f. Millivolts | Temperature, Degrees F. | Temperature Difference |
|---|---|---|---|---|---|
| −1.0 | −16.28 | 4.99 | 1.0 | 77.48 | 4.41 |
| −0.9 | −11.29 | 4.95 | 1.1 | 81.89 | 4.38 |
| −0.8 | − 6.34 | 4.91 | 1.2 | 86.27 | 4.36 |
| −0.7 | − 1.43 | 4.88 | 1.3 | 90.63 | 4.33 |
| −0.6 | 3.45 | 4.84 | 1.4 | 94.96 | 4.32 |
| −0.5 | 8.29 | 4.81 | 1.5 | 99.28 | 4.30 |
| −0.4 | 13.10 | 4.77 | 1.6 | 103.58 | 4.28 |
| −0.3 | 17.87 | 4.74 | 1.7 | 107.86 | 4.26 |
| −0.2 | 22.61 | 4.71 | 1.8 | 112.13 | 4.23 |
| −0.1 | 27.32 | 4.86 | 1.9 | 116.35 | 4.21 |
| 0.0 | 32.00 | 4.66 | 2.0 | 120.56 | 4.19 |
| 0.1 | 36.66 | 4.63 | 2.1 | 124.75 | 4.18 |
| 0.2 | 41.29 | 4.61 | 2.2 | 128.93 | 4.16 |
| 0.3 | 45.90 | 4.59 | 2.3 | 133.09 | 4.14 |
| 0.4 | 50.49 | 4.56 | 2.4 | 137.23 | 4.13 |
| 0.5 | 55.05 | 4.53 | 2.5 | 141.36 | 4.11 |
| 0.6 | 59.58 | 4.51 | 2.6 | 145.47 | 4.09 |
| 0.7 | 64.09 | 4.49 | 2.7 | 149.56 | 4.08 |
| 0.8 | 68.58 | 4.46 | 2.8 | 153.64 | 4.06 |
| 0.9 | 73.04 | 4.44 | 2.9 | 157.70 | 4.04 |
| 1.0 | 77.48 | | 3.0 | 161.74 | |

give an average e.m.f. provided all of the leads to the common junction points are of equal resistance. At times it is very desirable to have two or more couples arranged in parallel. The factors which may affect the reading of the couple are as follows, but they are usually smaller than the accuracy of most investigations:[4]

"1. Lack of homogeneity of the wires. This will introduce no variation unless it occurs in a portion of the wire in which there is a temper-

[4] P. Nichols, Temperature Measurements, American Society of Refrigerating Engineers, May, 1924.

ature gradient, as exists somewhere between the junctions. The effect is equivalent in action to having thermo-elements in the wire itself, and with a difference of temperature there will be an e.m.f. generated. Such an effect will not be large and can be tested for by changing the depth of insertion of the couple wires in the hot medium, or by otherwise creating temperature gradients at different points. Such stray e.m.f.'s are also caused by strain in the wires, and sharp bends should be avoided in the larger sizes, especially where they emerge from the hot or cold media.

" 2. Galvanic effects at any portion of the circuit. These will be reduced by orderly arrangement of the wires and by keeping them dry. It will be well to coat the insulation with paraffin.

" 3. E.m.f. from contact of dissimilar metals in the lead system. This may be in switches or lead wires. Care in locating instruments or switches so that different portions are not at different temperatures, will reduce this.

" 4. Caused by other electric circuits or by stray leakage. This can be tested for by opening and closing the switches in such lines to test as to whether the reading is different with or without those lines in use. Also by shielding the instruments—that is, mounting them on sheet metal plates, all of which are connected together.

"Too much importance cannot be paid to the proper insulating of the lead wires, terminal blocks and all parts of the circuits, especially if they are mixed up with high voltage used for heating or other test purposes. If the atmospheric conditions are good and the humidity low, there usually should not be any trouble; but in hot, humid weather, or damp working conditions, the stray leakage may become serious. The use of rubber covered copper leads is a good precaution and the ordinary No. 18 or 19 black fixture wire is very good in that it does not take on moisture at the surface so easily. The cotton covering over an ordinary flexible lamp cord may cause trouble, especially if the rubber is poor, and it is sometimes necessary to soak it in paraffin or cut it back for several inches at the ends, so as to expose the bare rubber.

" 5. Corrosion and deterioration, with proper care of the couples, are not factors at low temperatures."[5]

The wiring diagram for the Type K Leeds and Northrup potentiometer is shown in Fig. 216. This includes the potentiometer, lamp and

[5] It should be noted that where the galvanometer or the deflection method is used the reading is a function of the resistance of the leads, and any change in the resistance such as is produced by resistance, corrosion, or the switches, may seriously affect the calibration, while with the potentiometer the reading is independent of the resistance of the circuit.

scale, selector switch, standard cell galvanometer, battery or dry cell, and the necessary thermocouples and cold junctions. The manner of procedure is to check the battery against the standard cell by getting such a resistance in the potentiometer line that the battery current will just neutralize that of the standard cell and the net result will be zero deflection of the galvanometer. The battery now, for a short interval of time, will deliver a standard e.m.f. which tends to make a current of electricity flow in a certain direction. The e.m.f. of the couple tends to cause a current in the opposite direction and the resistance of the circuit is varied until the two currents are equal and the galvanometer deflection is zero.

**The Recording Thermometer.**—The recording thermometer may be used at times when results of approximate correctness are desired. For remote indication of temperature there are only three distinct methods employed:

*a*. The direct expansion of liquids.
*b*. The vapor tension of volatile liquids.
*c*. The expansion of inert gases.

The recording thermometer requires a bulb to be placed at the point where the temperature is required, and also a fine bore flexible tube connection to the indicating gage device attached to clockwork.

**Pressures.**—Pressures are usually given in pounds per square inch above the atmosphere, and are indicated either by a tube mechanism or by a diaphragm.

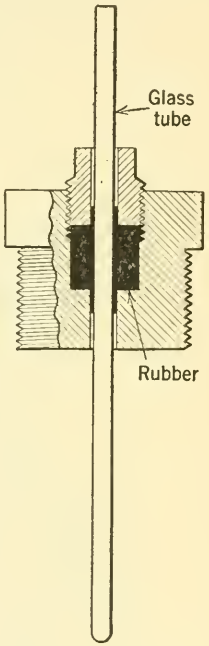


FIG. 219.—Thermocouple Fitting.

The tube for ammonia is of special steel, hardened and tempered, whereas for brine, steam, etc., other materials can be used. The tube is elliptical in cross-section and more or less semicircular in shape. As pressure is applied the tube tends to straighten and, by means of links, a rack, and pinion, a dial is made to turn. No gage of this sort should be used without calibration by means of a suitable dead weight tester. For accurate work, calibration should be done before, after and (for long tests) during the test. The barometer should be recorded at all times when accurate data are required as the ordinary gage indicates the pressure above or below the atmosphere.

Besides the absolute pressures there is frequent need of pressure

differences. Sometimes this can be given by means of a U tube containing mercury, oil, water or other liquids. Figure 221 gives a good design for obtaining differential pressures, such as would be needed in the

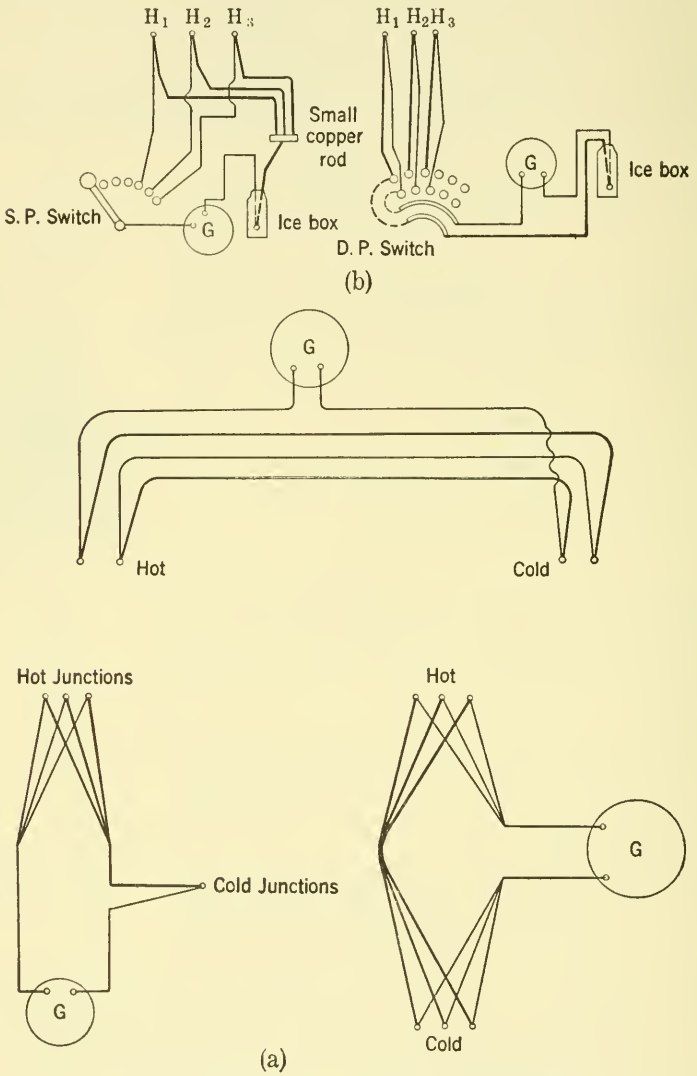



FIG. 220.—Thermocouple in Series and in Parallel.

ammonia gas line in order to show differences of pressure in the flow of superheated ammonia through pipes and fittings. The inverted U tube can be used in other cases, as, for example, in measuring the flow of

liquid ammonia through a thin plate orifice or Venturi meter; the differential height being in terms of the liquid flowing, which in this case is liquid ammonia. The float type differential gage,[6] designed for differential pressures up to 100 in. of mercury pressure, is shown in Fig. 222. This instrument can take static pressures up to 1000 lb. per sq. in.

**Measurement of Quantity.**—*Brine.*—If the brine is circulated by means of a centrifugal pump, then the Venturi and plate orifice method of measurement is satisfactory, but these methods are not accurate if

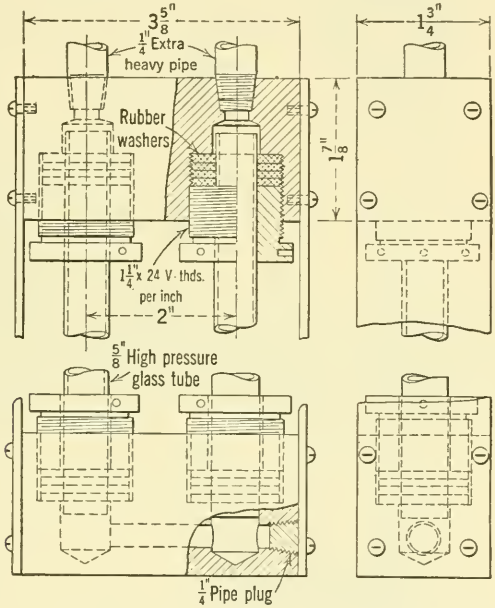



FIG. 221.—Differential U Tube Gage for Ammonia.

the reciprocating pump is used, on account of the pulsation developed. In any case the method of direct weighing on platform scales should be provided, or the orifice should be calibrated and then erected in place without change in the piping for a length equivalent to about 40 to 50 diameters of the pipe on both sides of the meter or orifice.

*Water.*—The amount of water passing over the condenser can be measured also by means of the thin plate orifice and the Venturi meter as well as the diaphragm piston and other forms of meters for measurement under pressure and by the rectangular and V notch weir. The principle of the fluid meter is as follows:

[6] The Foxboro Company.

**Venturi Meter.**

$$V_1 = A_1 \frac{1}{(R^4 - 1)^{\frac{1}{2}}} \times (2gh)^{\frac{1}{2}},$$

where

$V_1$ = the rate of flow in cubic feet per second;
$C$ = a constant = 0.97 to 1.00 as a rule;
$V_1 = M(2gh)^{\frac{1}{2}}$ theoretically;

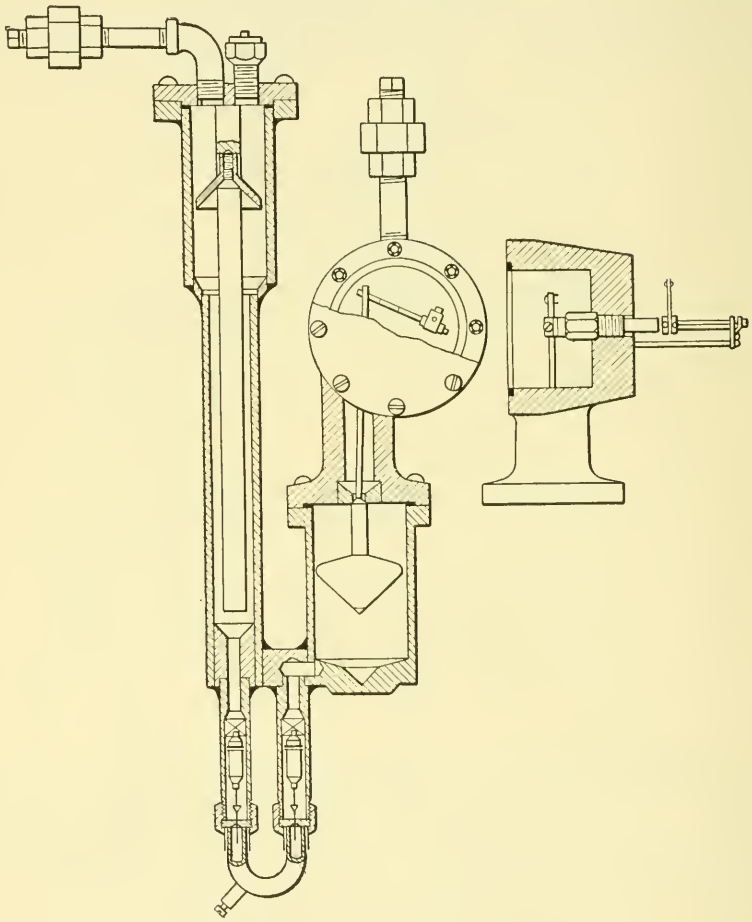


FIG. 222.—Differential Pressure Gage.

$$M = \frac{A_1}{(R^4 - 1)^{\frac{1}{2}}};$$

$V_1 = CM(2gh)^{\frac{1}{2}}$ practically,

$A_1$ = area entrance section at upstream connection, square feet.

$$R = \frac{D_1}{D_2},$$

where

$D_1$ = the entrance diameter;

$D_2$ = the throat diameter.

$C$ is also given as the value of $f\left(\frac{D_2 \times S_2 \times d}{u}\right)$,

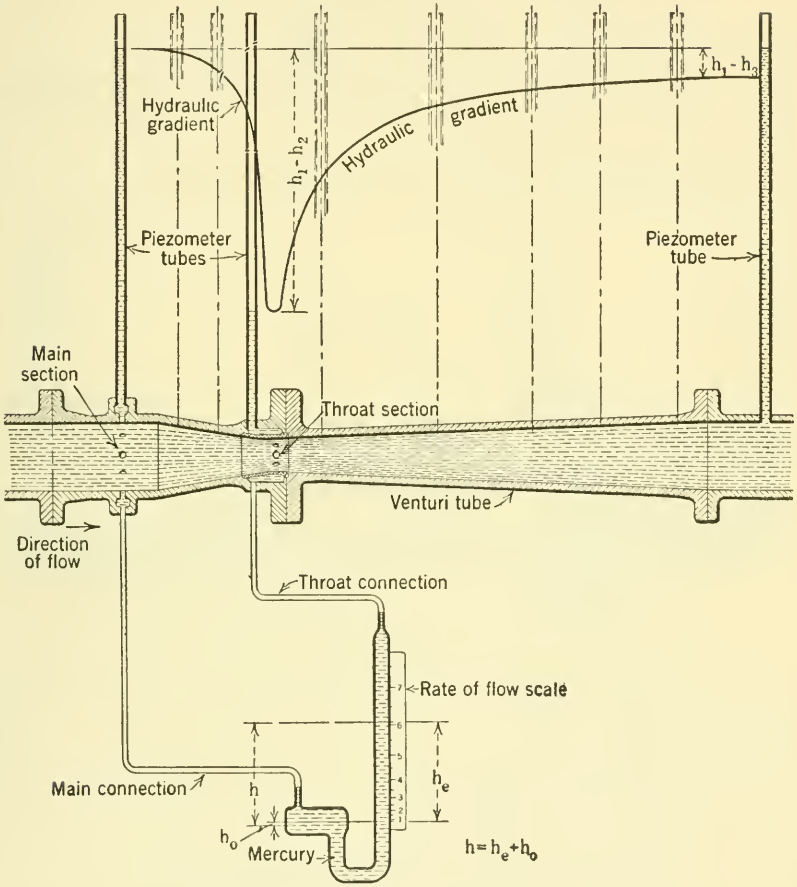



Fig. 223.—The Venturi Gage.

where

$S$ = the speed;

$d$ = the density;

$u$ = the viscosity;

$h$ is the differential level in feet of the liquid flowing.

**The Thin Plate.**—Here, also,

$$V = CM(2gh)^{\frac{1}{2}} \text{ cubic feet per second,}$$

where

$$M = \frac{A_1}{(R^4 - 1)^{\frac{1}{2}}};$$

$A$ = cross-section pipe, square feet;

$C$ = discharge coefficient = 0.60 to 0.62;

$R = \frac{D_1}{D_2}$, where $D_1$ = diameter of pipe, $D_2$ = diameter of orifice.

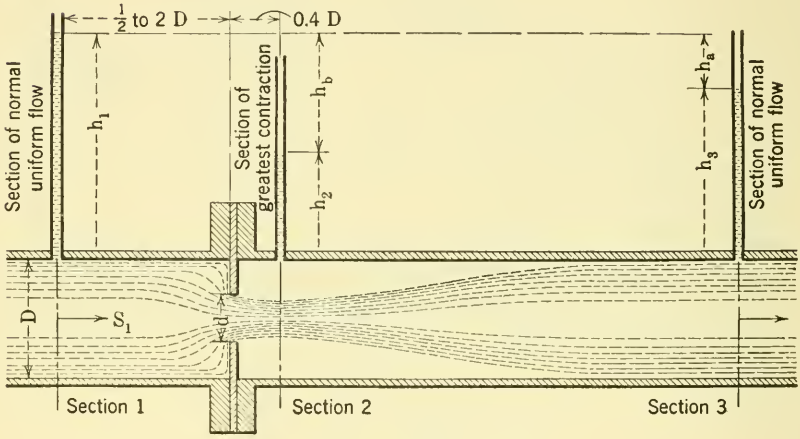



Fig. 224.—The Thin Plate Orifice.

If the differential pressure is not measured in feet of the liquid in the pipe but as a head $h_0$ (feet) of the liquid of density $d_0$, and $d$ = density of liquid to be measured, then

$$V = CM\left(2gh_0\frac{d_0}{d}\right)^{\frac{1}{2}},$$

and

$$W = CM(2gh_0d_0d)^{\frac{1}{2}} \text{ pounds per second.}$$

**Measurement of Refrigeration.**—Of all the quantities to be observed in testing refrigerating plants that of measuring the amount of the refrigeration is the most important, as this quantity is valuable in determining the heat balance of the condenser and the brine cooler and the performance of the compressor. The volume and the weighing methods have been used extensively for this purpose, especially for ammonia.

The weighing method is a satisfactory means of obtaining the amount of liquid refrigerant passing through the expansion valve. The apparatus consists of two drums on platform scales connected by means of flexible pipes (Fig. 225) with the header from the condensers. Flexible pipes for equalizing must be installed also, and the connection from the condenser should be very liberal in cross-section and should be erected with a constant slope to the weighing drums.

If the knife edges on the platform scales are in good shape, the weight of the liquid refrigerant can be obtained, usually, to within $\frac{1}{2}$ per cent error. The usual procedure is to weigh the drum empty and full and to alternate from one drum to the other. A test should be made to see what weight on the platform will make the beam rise through its full distance. Ammonia cocks can be used to advantage here so as to

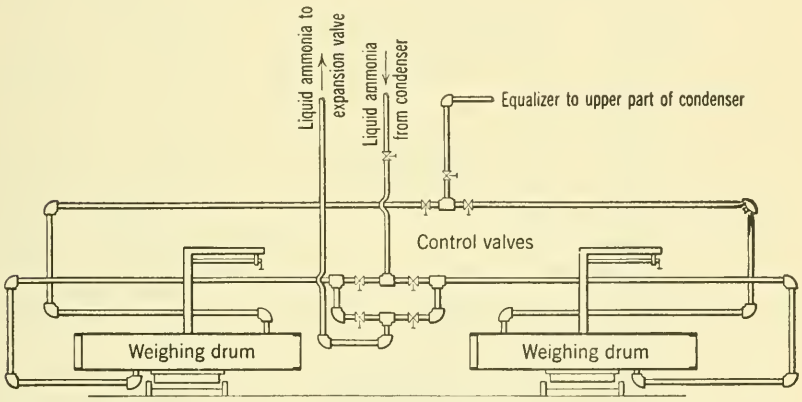



Fig. 225.—The Flexible Weighing Drum.

give a quick full opening and closing of the valve. Instead of weighing the total amount of ammonia passing through the expansion, the author prefers the method with the use of standard weights and an electric bell contact on the scale beam. In this case, with the drum normally empty, the standard weight is placed on the platform and the time noted with a stop watch when the beam makes contact at the top at which time the standard weight is removed. The time is again noted when the scale beam rings the bell again, as this will represent the lapse of time during the filling of the drum to an amount equal to the weight of the standard weight. Such a method requires *constant conditions* of operation, but no test is of any value unless the test conditions are constant. The test can be continued, alternating from one drum to the other, and the *rate of flow* can be secured in this manner. The total

weight recorded is not the *total* weight of condensate, but this total weight is not desired.

**The Calibrated Tanks for Ammonia.**—Where headroom will permit, the method of measuring the liquid ammonia by the use of calibrated

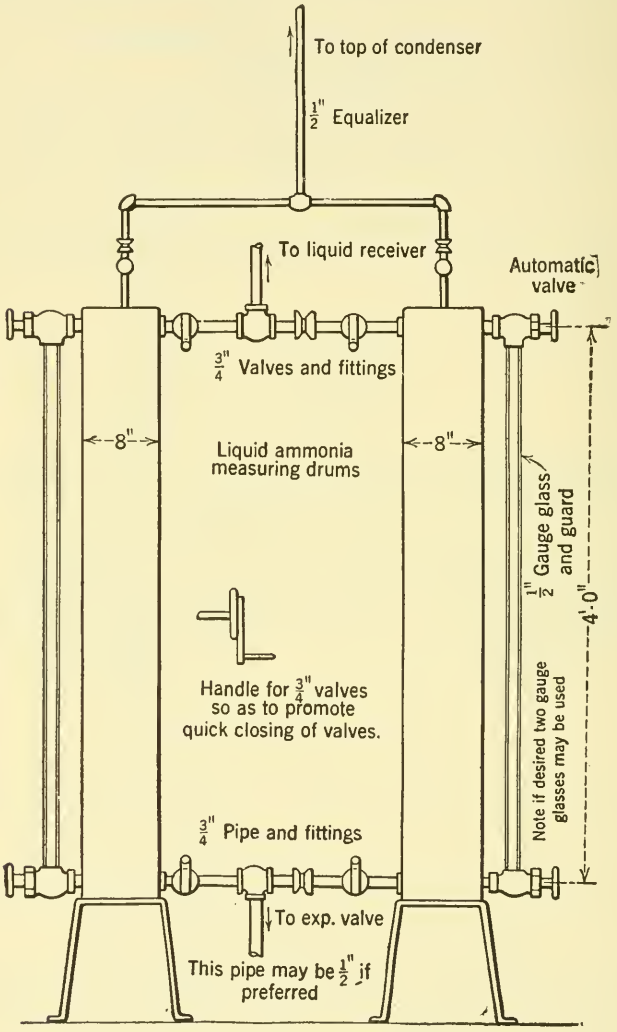

To top of condenser
1/2" Equalizer
To liquid receiver
Automatic valve
3/4" Valves and fittings
8"
8"
Liquid ammonia measuring drums
1/2" Gauge glass and guard
4'-0"
Handle for 3/4" valves so as to promote quick closing of valves.
Note if desired two gauge glasses may be used
3/4" Pipe and fittings
To exp. valve
This pipe may be 1/2" if preferred

Fig. 226.—The Calibrated Drum.

tanks is sometimes better than by the use of the weighing drums (Fig. 226). In this case, also, cocks are better than valves if the cocks can be made tight, but tightness is the first essential. The upper connection

to the stand of condensers should be liberal, and each tank should be fitted with a gage glass of the best quality glass and with automatic stop valves. The calibration should be made after everything is piped ready for operation, by syphoning in water of a known weight and temperature, for which a scale of cubic feet can be constructed.

The particular advantage of the calibrated tank is that a check reading can be made readily. For example, operating conditions are established and maintained for a certain interval of time. After that it would be expected that the amount of liquid leaving one drum would always equal the amount entering the other. If this is not so then equilibrium has not been established, and the irregularity should be explained. As vertical tanks only are approved, a check can be made of the *rate of flow* by readings at frequent time intervals. As the tank calibration is in cubic feet the temperature of the liquid in the tank needs to be taken in order to reduce to pounds.

Both the weighing and the volume methods of measuring the liquid refrigerant need cumbersome apparatus, not easily moved from one plant to another like an engine indicator or a test wattmeter. Attempts have been made[7] to demonstrate the value of the Venturi meter for measuring liquid ammonia. While the Venturi meter and the flat plate orifice are excellent means of measuring water and brine, yet with liquid ammonia the case is different, for in this case there is a volatile fluid tending to form bubbles of gas with a decrease of pressure or an increase of temperature. For example, if the air temperature is 90 deg. F. and the condenser pressure is 150 lb. gage (corresponding to a temperature of saturation of 84.4 deg.), then there will be a tendency to absorb heat and vaporize. A drop of pressure will take place in the meter which will cause trouble also. The method advocated is to precool the liquid ammonia before it arrives at the Venturi (or thin plate orifice) by means of the coldest water available at the plant. The flow meter can be used to advantage to adjust the expansion valve, but it has not been proved to be a reliable means of securing the weight of the refrigerant.

**Indicating the Power of the Compressor.**—The valves of the ammonia compressor are not mechanically operated[8] and therefore they cannot be set improperly. The suction valve of the vertical compressor is usually a balanced valve, while the horizontal compressor can have a number of designs, and frequently employs the plate (ring plate) type. The indicator diagram is of no value in showing the setting of the valves, although it can show sluggish action, excess pressure in the cylinder due to inertia of the valve, excess pressure loss in getting the

[7] Compound Compressors, Geo. Horne, Refrigerating Engineering, 1923.

[8] The sleeve valve of the H. J. West compressor *is* mechanically operated (Fig. 47).

gas out of the cylinder and into the condenser and leaky valves and piston rings.

In using the steam engine indicator very great care must be exercised, especially if the speed is over 100 r.p.m., or inaccurate results will be experienced.[9] The principal sources of error are:

*a.* Inaccurate springs.

*b.* Excessive friction in the indicator on account of the piston.

*c.* Loss motion, poor design and excess inertia in the indicator mechanism.

*d.* Inaccurate reducing motion, inertia effects of the drum, stretching of the indicator cord.

The outside spring indicator has the advantage of the inside spring type because of the lesser temperature variation in the indicator cylinder, although this is less important in ammonia compression than in steam engine testing.

**Reducing Motion.**—An indicator card about $3\frac{1}{2}$ in. long is usually desired, and some connection is necessary to the crosshead or the shaft through a special mechanism in order to secure this reduced motion. If the crosshead is conveniently located some form of pantagraph is usually the best, or a specially designed reducing motion can be laid out on the drawing board for the particular compressor. Whatever type of reduction is used should be centered accurately, the pins should be of steel, hardened and ground, and arranged for lubrication. If the crosshead cannot be reached, an eccentric on the shaft or a crank and connecting rod should be used, designed for the *same* ratio of crank to connecting rod as in the compressor. The reduced motion can be extended to a convenient point, where attachments to the indicator can be made, by means of steel wire kept taut all the time by means of a helical spring.

For speeds of 200 r.p.m. and over the steam engine indicator is of doubtful value. Under such conditions some other form of indicator, the optical or the Bureau of Standards' diaphragm type, for example, should be used.

**Other Test Observations.**—There should be means of measuring the speed, the electrical power input (if electric driven), the density and the specific heat of the brine, and of the condensing water if this is not fresh water, the barometer, etc. If the compressor is steam-engine driven, indicator cards of the steam cylinder, the weight of the steam and the initial quality of the steam are desired. A test code for refrigerating machines has been prepared jointly by the American Societies of Mechanical and Refrigerating Engineers, and is given in the following:

[9] Experimental Engineering, Carpenter and Dieterichs, John Wiley and Sons.

### Test Code for Refrigerating Systems

1. The Test Code for Refrigerating Systems is intended for use in the determination of the performance of compression systems in which compressors of the reciprocating type are used as well as absorption machines. The term "refrigerating system," as used in this Code, includes all the necessary machinery and apparatus directly used in the refrigerating cycle for determining the refrigerating effect, power consumption, and other primary data, indicated in the following tables.

It does not include ice making and other apparatus employed in the utilization of refrigeration. In so far as the fundamental operations of all the systems are in common, general rules will be laid down and where the systems differ separate rules will be given. For the test of the driving element in compression systems, the A.S.M.E. Power Test Codes for Steam Engines, Steam Turbines, Internal-combustion Engines, etc., should be consulted for details not contained in this Code.

### Object

2. In accordance with the "Code on General Instructions," the object of the test should be first determined and recorded. If the object relates to the fulfilment of a contract guarantee, an agreement should be made between the interested parties concerning all matters about which disputes may arise (see "Code on General Instructions," Par. 6), and a full statement of the agreement entered in the report of the test.

### Measurements

3. The principal measurements and quantities determined in a test are:

(*a*) The quantity of the refrigerant circulated;
(*b*) The various temperatures of the refrigerant in the cycle;
(*c*) The various pressures of the refrigerant in the cycle;
(*d*) The energy required to operate the driving element and the various auxiliary apparatus;
(*e*) The various temperatures of condensing and cooling water;
(*f*) The quanity of condensing and cooling water;
(*g*) Quantity, quality and pressure of steam used in the steam cylinder or generator;
(*h*) Principal dimensions of driving element (see appropriate code);
(*i*) Quantity of composition and density of brine used;
(*j*) Temperature of brine entering and leaving the evaporator;
(*k*) Quality of liquor used in absorption plants.

### Instruments and Apparatus

4. The instruments and apparatus necessary in carrying out the test are:

(*a*) Suitable meter for measuring liquid refrigerant on the high- or intermediate-pressure side where it passes to the expansion coils of the brine cooler or elsewhere;
(*b*) Platform scales and suitable tanks for measuring water and condensed steam;
(*c*) Water meters, calibrated tanks or tank and platform scale for measuring, condensing and cooling water;

(*d*) Pressure gages and vacuum manometers, including a suitable U tube or mercury column to be connected at the low-pressure suction of the compressor. It is recommended that this be used in all cases rather than a pressure or vacuum gage for suction pressure under 20 lb. gage;
(*e*) Calibrated thermometers and thermometer mercury wells of sufficient lengths to extend beyond the centers of the pipes. Thermometers used for taking temperatures of brine shall be graduated in 1/10 deg. F. and readings estimated to 1/100 deg. F.;
(*f*) Barometer and psychrometer;
(*g*) Revolution counter or other accurate speed measuring device;
(*h*) Graduated stroke indicator for direct-acting pumps;
(*i*) Pycnometer, Mohr-Westphal balance, or calibrated hydrometer for specific gravity of brine;
(*j*) Calorimeter for determining the quality of the steam;
(*k*) Steam and ammonia indicator;
(*l*) Planimeter;
(*m*) Appropriate instruments for measuring power consumption of motor-driven machines.

The direction for the use, calibration and accuracy of the instruments and apparatus enumerated above are given in the various sections of the "Code on Instruments and Apparatus."

### Preparation

5. Paragraphs 7 to 11 of the "Code on General Instructions" should be read, carefully studied and conformed to wherever they apply. The dimensions and the physical conditions of all parts of the plant essential to the object of the test should be accurately determined and carefully recorded.

6. The dimensions of the compressor and engine cylinder and the pump plungers or cylinders including the volume of the clearance should be accurately measured and recorded in accordance with Par. 6 of the A.S.M.E. "Test Code for Reciprocating Steam Engines," and Par. 12 of the A.S.M.E. "Test Code for Displacement Compressors and Blowers."

7. All measuring devices should be installed and in the case of the meters for measuring water or brine a means of calibrating by direct weighing or checking by calibrated tanks should be provided. The meter for measuring the liquid refrigerant shall be calibrated by means of a closed tank or receiver of known volume. This volume should preferably be determined by weighing the water which fills the tank or receiver. In the ordinary plant a liquid receiver may be used for this purpose.

8. The thermometer wells should be of steel for ammonia lines, and bronze for brine lines, and should extend into the pipe a sufficient distance so that the bulb of the thermometer will extend at least to the center of the pipe. When comparisons of temperatures are being made it is extremely important that the immersion in the well should be identical in all cases. These wells should be filled with mercury to the point indicated on the stem of the thermometer. Insulation on the cold lines should extend even to the top of the well, and the thermometers pass through corks fitted to the wells and bored to accommodate the thermometers. The thermometer well must be placed at least 10 ft. away from brine coolers or pumps so that the solution has uniform temperature where the readings are taken. Electric resistance

thermometers or thermocouples may be used in taking temperatures of brine as a check on mercury thermometers.

9. In testing a plant where the load is light a brine heater or a heater using waste condensing water flowing over the evaporating coils should be used. An agreement should be reached as to which of the interested parties is to supply and install the brine heater, if necessary, and all other apparatus required for the tests.

10. For ammonia a special steel indicator must be provided which has steel indicator cocks with short separate connections to the ends. The indicator should be provided with a stop so that a light spring may be used for studying suction conditions.

11. In ammonia absorption systems the anhydrous ammonia should not contain over 3 per cent moisture. Samples should be drawn before and immediately after the test and determinations of moisture made.

### Operating Conditions

12. The operating conditions should conform to the object of the test and should prevail throughout the trial as pointed out in Par. 23 of the "Code on General Instructions." Uniformity of conditions and evenness of operation must prevail throughout the test. It is required under this Code that nothing but liquid shall enter the expansion valve and nothing but vapor shall enter the refrigerating machine. In commercial tests the allowable percentage of difference in the heat balance between the water and the refrigerant must be stated.

### Starting and Stopping

13. The plant should be operated a sufficient length of time prior to the starting of the test to insure uniformity of conditions and, when these conditions are obtained, the test should be started and continued as stated in Par. 14 of this Code. It is essential to the accuracy of the test that all parts of the plant contain the same amount of heat estimated above some datum at the end as at the beginning of the test. To accomplish this all vessels containing liquid refrigerant should be supplied with suitable gage glasses so that the same quantity of fluid may be distributed alike at the beginning and end of the test and the quantities and temperature of the fluids in the various parts maintained as uniform as possible.

### Duration

14. Each run shall be continued over a period of time not less than $t$ hours, $t$ to be calculated from the following formula:

$$tw_a = 3w_b$$

where $w_a$ = approximate weight of anhydrous refrigerant circulated in pounds per hour, and

$w_b$ = total weight of anhydrous refrigerant contained in the system.

### Records

15. The data should be taken and recorded in the manner prescribed in the "Code on General Instructions," Pars. 24 to 35. Fifteen minute readings will be sufficient, except where there is considerable fluctuation in the readings, in which case more frequent readings must be taken to insure good averages.

16. Each indicator card should be marked with the date, time of day, cylinder and end of cylinder, and identification mark and strength of indicator spring, as pointed out in the "Test Code for Reciprocating Steam Engines," Par. 20. To assist in the uniformity of operation a chart should be plotted during the test of the principal quantities, including the estimated tonnage, as pointed out in Par. 39 of the "General Instructions."

17. The data and results should be tabulated in accordance with the form shown in the following tables, adding items not provided for and omitting items not needed to conform to the object in view. Unless otherwise indicated the items refer to the numerical readings which are recorded in the log.

## Calculation of Results

18. *Standard Ton of Refrigeration.*—The now generally adopted recommendations of the Joint Committee of The American Society of Mechanical Engineers and American Society of Refrigerating Engineers on Standard Tonnage Basis for Refrigeration are as follows:

(1) A standard ton of refrigeration is 288,000 B.t.u.

(2) The standard commercial ton of refrigeration is at the rate of 200 B.t.u. per minute.

(3) The standard rating of a refrigerating machine, [1] using liquefiable vapor, is the number of standard commercial tons of refrigeration it performs under adopted refrigerant pressures. [2]

[1] Note 1. A refrigerating machine is the compressor cylinder of the compression refrigerating system, or the absorber, liquor pump, and generator of the absorption refrigerating system.

Note 2. Nothing but liquid shall enter the expansion valve, and nothing but vapor shall enter the refrigerating machine.

Note 3. There shall be 9 deg. F. (5 deg. C.) sub-cooling of the liquid entering the expansion valve and 9 deg. F. (5 deg. C.) superheating of the vapor entering the refrigerating machine, the points at which the sub-cooling and the superheating are determined to be within 10 ft. of the expansion valve and refrigerating machine, respectively.

[2] Note 4. The inlet pressure is that which corresponds to a saturation temperature of 5 deg. F. (−15 deg. C.).

Note 5. The outlet pressure is that which corresponds to a saturation temperature of 86 deg. F. (30 deg. C.).

Note 6. These pressures are measured outside and within 10 ft. of the refrigerating machine, distances which are measured along the inlet and outlet pipes, respectively.

19. The refrigerating output of the system shall be expressed in standard commercial tons of refrigeration as calculated from the weight of liquid refrigerant and the amount of available cooling effect produced in the evaporator. This method shall be employed in the case of all refrigerants where the physical properties have been determined and where tables of such properties are recognized as sufficiently accurate. The tables of thermodynamic properties employed shall be those adopted and published by The American Society of Refrigerating Engineers.

20. The weight of refrigerant circulated shall be determined by a suitable liquid meter as prescribed in the A.S.M.E. Code on Instruments and Apparatus. As an

alternate method the weight of liquid refrigerant may be calculated from the volume as obtained from a pair of calibrated receivers or by direct weighing.

21. The amount of available refrigerating effect shall be the weight of refrigerant circulated per hour multiplied by the difference in heat content on entering and leaving the evaporator. Where there is more than one evaporator in which different pressures exist, the weight of refrigerant to each evaporator and the total heat from the vapor must be determined separately and the total refrigerating effect may then be computed by the following formula:

$$Q_e = w_a(h_1 - h_4)$$

where $Q_e$ = total refrigerating effect per hour;
$w_a$ = weight of anhydrous refrigerant circulated in pounds per hour;
$h_1$ = total heat of the vapor leaving the evaporator;
$h_4$ = total heat of the liquid at the expansion valve.

22. The refrigeration effect expressed in standard commercial tons of refrigeration is

$$R.E. = \frac{Q_e}{12{,}000}$$

23. *Indicated Horse Power.*—The indicated horse power for each end of the cylinder is found by using the formula:

$$\text{i.hp.} = \frac{p_m LAN}{33{,}000},$$

where $p_m$ = mean effective pressure in pounds per square inch;
$L$ = length of the stroke in feet;
$A$ = area in square inches of the piston less the area of the piston rod, if any;
$N$ = number of revolutions per minute.

The total horse power of the cylinder is the sum of the horse power developed in the two ends.

24. *Mean Effective Pressure.*—The mean effective pressure should be found by dividing the area of the indicator diagram in square inches as determined with a planimeter by the length of the diagram in inches, and multiplying the quotient by the scale of the indicator spring. If a planimeter is not available, the approximate mean effective pressure may be determined by finding the average height of the diagram in inches as obtained by averaging a suitable number of equally spaced ordinates, at least twenty, measured between the lines of the forward and return strokes, and then multiplying this average by the scale of the spring. The length of the indicator diagram is the measured distance along the atmospheric-pressure line between ordinates erected perpendicular to it and passing through the ends of the indicator diagram.

25. *Theoretical Horse Power.*—The theoretical horse power required to compass adiabatically the quantity of vapor shall be found in the case of ammonia or fluids having standard tables or charts, by referring to the Mollier Chart or the tables adopted by The American Society of Refrigerating Engineers.

$$hp = \frac{(h_1 - h_2) \times w_a \times 778}{33{,}000 \times 60},$$

where

$h_1$ = heat content of vapor at compressor inlet;
$h_2$ = heat content of vapor after adiabatic compression;
$w_a$ = weight of fluid circulated, pounds per hour.

26. In the case of fluids not having published charts or tables, the following method may be used to calculate the theoretical horse power required to compress the vapor adiabatically at the average rate for a single stage or for any stage of mutiple-stage compressors:

$$hp = \frac{144k}{33{,}000(k-1)} p_1 v_1 \left[ \left( \frac{p_2}{p_1} \right)^{\frac{k-1}{k}} - 1 \right],$$

where $v_1$ = volume compressed, cubic feet per minute;
$p_1$ = absolute intake pressure, pounds per square inch;
$p_2$ = absolute discharge pressure, pounds per square inch;
$k$ = the exponent in $pv^k$ = constant.

27. *Gross Horse Power.*—The gross horse power is the indicated horse power in the steam of power cylinders in the case of a steam or internal combustion engine-drive compressor; the electrical horse power multiplied by the motor efficiency in the case of a direct connected motor-driven compressor; and the brake horse power delivered to the compressor shaft in the case of a belt-driven or gear-driven compressor. If motor efficiency determined under the A.I.E.E. rules is used, the value obtained will be a conventional one, and will always be higher than the actual efficiency of the motor. An agreement must therefore be made by those interested in the test as to the tolerance to be applied to the "conventional efficiency" to convert the figure into actual efficiency.

28. *Electrical Horse Power.*—The electrical horse power of a motor is found by dividing the input at the terminals expressed in kilowatts by the constant 0.746. In the case of an alternating-current motor, the input determined, whether expressed in electrical horse power or kilowatts, should be the total input. When the power for excitation is taken directly from the compressor shaft, the total input is that indicated at the a.c. motor terminals. When the power for excitation is furnished from an independent source the total electrical horse power used in calculating the gross horse power (Par. 27) is that indicated as the motor terminals plus an equivalent power required to run the exciter.

29. *Volumetric Efficiency.*—The volumetric efficiency is the ratio of the capacity of the compressor to displacement. The capacity is the actual amount of vapor compressed and delivered, expressed in cubic feet per minute at intake temperature and pressure.

30. *Mechanical Efficiency of the Compressor.*—The mechanical efficiency is the ratio of the indicated horse power of the compressor cylinder to the indicated horse power in the power cylinders in the case of a steam-driven or internal combustion engine-driven compressor, and to the brake horse power delivered to the shaft in the case of a power-driven machine..

31. *Compressor Efficiency.*—The compression efficiency in a single-stage compressor is the ratio of the horse power required to compress adiabatically all the vapor delivered by the compressor to the horse power developed in the compressor cylinder, as shown by the indicator cards. The compression efficiency for any par-

ticular cylinder of a multi-stage compressor is the ratio of the horse power required to compress adiabatically all of the vapor delivered by the compressor through the observed pressure range for the particular cylinder in question to the horse power developed in the particular cylinder in question as shown by the indicator cards. The two factors involved in this ratio are defined in Pars. 23 and 25, respectively.

32. *Condenser Performance.*—The total heat removed by the condensing water shall be calculated from the observed data when the enclosed type of condensers, either shell and tube or double pipe, are used. The total heat may also be calculated from the quantity of refrigerant circulated. In this case

$$Q_e = w_a(h_3 - h_4),$$

where $Q_e$ = heat removed per hour;
$h_3$ = heat content of vapor entering condenser;
$h_4$ = heat content of fluid leaving condenser;
$w_a$ = weight of refrigerant condensed per hour.

33. *Determination of Heat Transfer in Condensers.*—The logarithmic mean difference is based upon a constant temperature of liquefaction throughout the condenser on the assumption that the heat transfer is uniform throughout the condenser. While the calculation of heat transfer per square foot of refrigerant surface per hour per degree of logarithmic mean difference does not give entirely satisfactory results, the determination of heat transfer shall be calculated in this manner.

$$\text{Log. mean difference} = \frac{t_2 - t_1}{\log_e \dfrac{t_e - t_1}{t_e - t_2}},$$

where $t_e$ = temperature of liquefaction corresponding to condenser pressure;
$t_1$ = temperature incoming water;
$t_2$ = temperature outgoing water.

If a complete condenser test is to be made the A.S.M.E. Test Code for Condensing Apparatus and the Code on Instruments and Apparatus should be carefully studied.

34. *Evaporator Performance.*—The evaporator performance shall be calculated from the observed data. In this case

$$Q_e = w_a(h_5 - h_6),$$

where $Q_e$ = heat removed per hour;
$h_5$ = heat content of vapor, leaving evaporator;
$h_6$ = heat content of liquid entering expansion valve;
$w_e$ = weight of refrigerant entering evaporator in pounds per hour.

35. *Determination of Heat Transfer in Evaporator.* (*a*) Brine Coolers.—Brine cooler performance shall be expressed in terms of heat transfer per square foot of cooling surface per hour per degree of logarithmic mean difference.

$$\text{Log. mean difference} = \frac{t_4 - t_3}{\log_e \dfrac{t_4 - t_5}{t_3 - t_5}},$$

where $t_3$ = temperature of brine out of evaporator;
$t_4$ = temperature of brine to evaporator;
$t_5$ = temperature of vaporization corresponding to evaporator pressure.

(*b*) The performance of ice tanks and all other forms of expansion coils or apparatus will not be specifically covered in this Code, which is intended to include the main items of the refrigerating cycle rather than the products of the cycle.

36. *Water Horse Power.*—The water horse power output at observed total suction and discharge pressure is

$$\text{w.hp.} = \frac{(\text{lb. of liquid per minute}) \times (\text{total head in ft.})}{33{,}000}.$$

The gross or brake horse power input shall be observed for all pump-driving elements and the efficiencies calculated.

37. *Heat Balance.*—The general formulas for heat balance are as follows:

$$\text{Compression Cycle: } Q_e + Q_w = Q_1 + Q_3;$$

$$\text{Absorption Cycle: } Q_e + Q_s = Q_1 + Q_2 + Q_3,$$

where $Q_e$ = heat absorbed by evaporating refrigerant;
$Q_w$ = heat equivalent of work in compressor;
$Q_s$ = heat imparted by steam in generator;
$Q_1$ = heat rejected in condenser;
$Q_2$ = heat rejected in absorber;
$Q_3$ = heat rejected or radiated in addition to $Q_1$ and $Q_2$.

38. For purposes of illustration the following list of quantities involved in the computation of the heat balances of compound compression systems is given:

HEAT ABSORBED (B.t.u. per hour)

(*a*) Heat absorbed in evaporator;
(*b*) Heat entering evaporator insulation;
(*c*) Heat absorbed in low-pressure suction main;
(*d*) Heat absorbed in low-pressure suction trap;
(*e*) Heat equivalent of work done in compressor;
(*f*) Heat absorbed from engine room through cold surface of low-pressure compressor;
(*g*) Heat absorbed through surface of intermediate liquid receiver;
(*h*) Heat absorbed through surface of intermediate-pressure liquid line;
(*i*) Heat absorbed from engine room through surface of high-pressure suction main;
(*j*) Heat absorbed through cold surfaces of high-pressure compressor;
(*k*) Heat absorbed or rejected through condenser shells.
(*l*) Heat absorbed or rejected through receivers.
(*m*) Heat absorbed or rejected in high-pressure liquid line.

HEAT REJECTED (B.t.u. per hour)

(*n*) Heat rejected by hot surface of low-pressure compressor;
(*o*) Heat rejected from low-pressure discharge main between low-pressure compressor and intermediate vapor cooler;
(*p*) Heat rejected in intermediate vapor cooler;
(*q*) Heat rejected to engine room by intermediate vapor cooler;
(*r*) Heat rejected in discharge main from intermediate vapor cooler to intermediate liquid receiver;
(*s*) Heat rejected by hot surfaces of high-pressure compressor;
(*t*) Heat rejected by high-pressure discharge main and oil separator between machine and condensers;
(*u*) Heat rejected in ammonia condensers;
(*v*) Heat rejected in liquid cooler;
(*k*) Heat rejected or absorbed through condenser shells;
(*l*) Heat rejected or absorbed through receivers;
(*m*) Heat rejected or absorbed in high-pressure liquid line.

Before concluding tests, and making a final report, it is recommended that Par. 20 of the "Code on General Instructions" should be carefully read. The data and results should be reported in accordance with the forms prescribed by the committee.

## CHAPTER XII

## PIPING

A large amount of the equipment in the compression, and very nearly all of it in the absorption, refrigerating plant is *piping*. In some special cases, as, for example, the plant using the shell and tube condenser, the shell and tube brine cooler and, in the extreme case, a brine spray system of cooling air, the amount of piping would be comparatively small. In most cases, however, both the suction and lines should be calculated carefully, as well as the condenser headers and liquid drains and the piping on the low pressure side, or the brine piping if brine is used. Like all other branches of engineering, the piping must be efficient as regards operation and overhead costs, it must be arranged for draining, cleaning, pumping out and repairs, and it must be safe and self-contained. These considerations will be taken up in the following chapter.

**Kind of Pipe.**—Full weight steel pipe is quite generally used for ammonia at the present time, even for condensers and ice tanks, whereas carbon dioxide refrigeration uses extra heavy steel pipe, because of the much heavier unit pressures. Wrought-iron pipe is not so commonly used as it was some years ago, because it has been found that steel pipe can be depended on much better than formerly, both as regards the welded joint and the thickness of the walls. It is usual to specify butt-welded pipe for sizes 2 in. and smaller, but lap-welded pipe (drawn down from a larger size) for normal diameters larger than 2 in.

**Piping at the Compressor.**—The headers and connections at the compressors are usually irregular in design and therefore are frequently welded, although the cast iron or cast semi-steel bends are used whenever the cost of the casting is less than the welded header. Considerable care should be taken with the piping to eliminate changes in the direction of the gas flow as far as possible, since it has been determined that irregular flow through the cylinder ports (with sharp changes of direction of flow) and the suction and discharge bends is much more likely to cause loss of pressure of the gas than is the high velocity of the gas flowing through the valves due to small cross-sectional area. The discharge port areas sometimes are designed for 10,000 ft. per min. gas velocity, but if this is so the ports should be straight and when changes

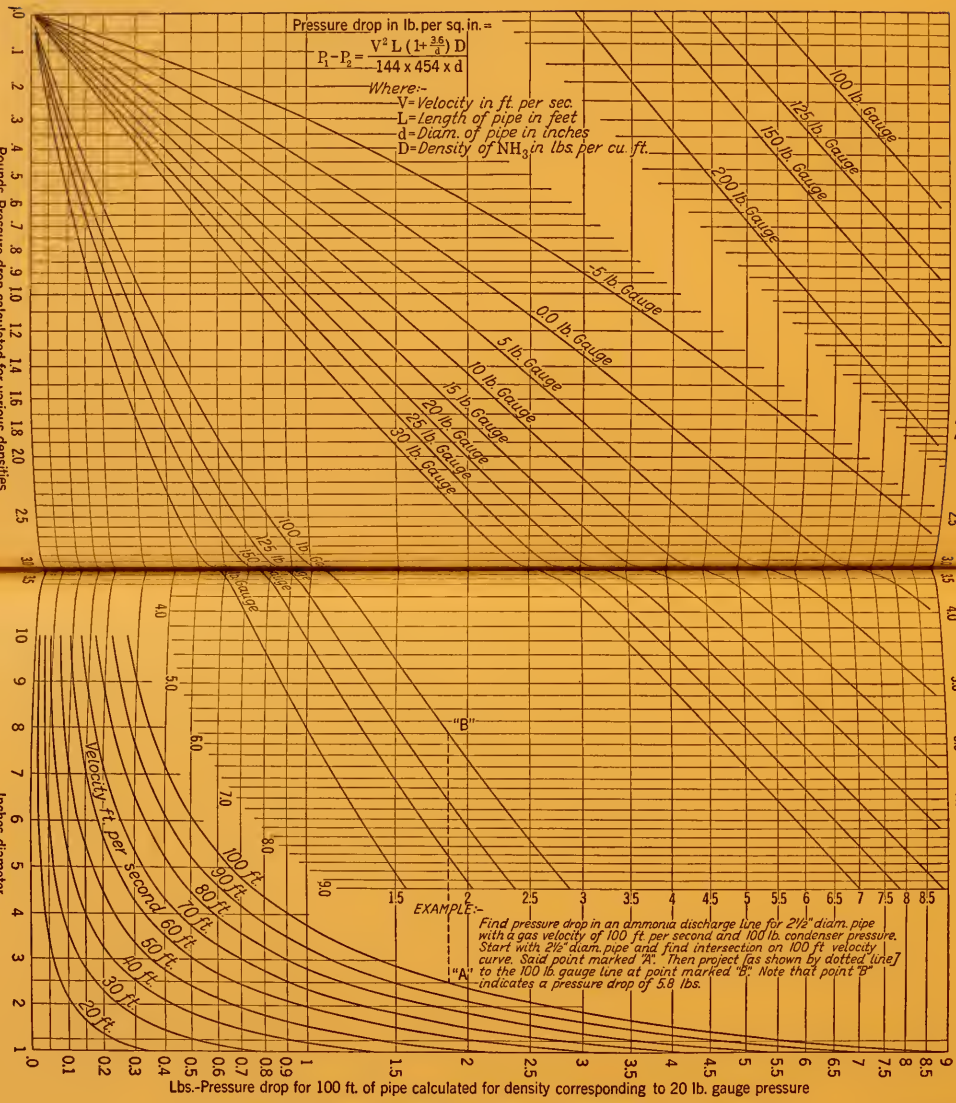

Pressure drop in lb. per sq. in. =
$P_1 - P_2 = \frac{V^2 L (1 + \frac{3.6}{d}) D}{144 \times 454 \times d}$
Where:-
V=Velocity in ft. per sec.
L=Length of pipe in feet
d=Diam. of pipe in inches
D=Density of $NH_3$ in lbs. per cu. ft.
-5 lb. Gauge
0.0 lb. Gauge
5 lb. Gauge
10 lb. Gauge
15 lb. Gauge
20 lb. Gauge
25 lb. Gauge
30 lb. Gauge
100 lb. Gauge
125 lb. Gauge
150 lb. Gauge
200 lb. Gauge
Pounds-Pressure drop calculated for various densities
Lbs.-Pressure drop
Velocity ft. per second
100 ft.
90 ft.
80 ft.
70 ft.
60 ft.
50 ft.
40 ft.
30 ft.
20 ft.
Inches diameter
"B"
"A"
EXAMPLE:-
Find pressure drop in an ammonia discharge line for 2½" diam. pipe with a gas velocity of 100 ft. per second and 100 lb. condenser pressure. Start with 2½" diam. pipe and find intersection on 100 ft. velocity curve. Said point marked "A". Then project [as shown by dotted line] to the 100 lb. gauge line at point marked "B". Note that point "B" indicates a pressure drop of 5.8 lbs.
Lbs.-Pressure drop for 100 ft. of pipe calculated for density corresponding to 20 lb. gauge pressure

FIG. 227.—Loss of Pressure Due to Flow of Ammonia in Pipe Lines.

*To face page 355.*

of direction are necessary they should be accomplished by means of long radius bends.

**Size of the Discharge Pipe.**—Refrigerating engineers have not paid much attention to the discharge ports, passages, and the diameter of the piping, but if the compressor horse power is to be reduced to a minimum value the subject of the pressure drop from the cylinder to the condenser must be gone into carefully. It has been the custom to calculate the *average* gas velocity in the discharge from the expression:

$$\text{Average gas velocity} = \frac{\text{piston area} \times \text{stroke} \times 2 \times \text{r.p.m.}}{\text{cross-sectional area of the pipe}}$$

$$= 6000 \text{ to } 7000 \text{ ft. per min.},$$

if the compressor is double-acting or if it is a twin cylinder, single-acting machine. This calculated value is really only an average gas velocity, whereas the actual velocity is a maximum near mid-stroke and is equal to zero at the dead centers. The result of the intermittent discharge is that the gas pressure is built up in the discharge line and to a lesser extent in the condenser as these volumes act in a manner similar to the air chamber on the discharge line of pumps and compressors. The excess pressure, over and above that of the condenser pressure, shown on many indicator diagrams, is due to the friction to flow and the momentary discharge into a chamber of constant volume.

**Size of the Suction Line.**—The size of the suction pipe line is very much more important than that of the discharge line. If undue losses of pressure occur in the discharge line the effect will be a decreased volumetric efficiency of the compressor as well as an increased work of compression. However, wire-drawing in the suction line drops the capacity of the compressor rapidly (Fig. 29), increases the horse power per ton of refrigeration and decreases the volumetric efficiency. The results are that, if necessary, the discharge line may be designed for high average gas velocities, but it does not pay to permit velocities greater than 5000 ft. per min. in the suction, and long lines may be as low as 2000 ft. per min., with an average of 3000 ft. An idea can be secured of the loss of pressure from Fig. 227 and the following problem:

**Problem.**—A 2-in. pipe line for a suction pressure of 20 lb. gage is 300 ft. long. Find the pressure drop if the gas velocity is 5000 ft. per min. Referring to the figure it will be seen that the drop per 100 ft. is 1.85 lb., and the total drop will be $3 \times 1.85$ or 5.55 lb.

**Condenser Piping.**—The details of condenser piping depend on the kind of condenser, some of which are shown in Figs. 228 to 230. When

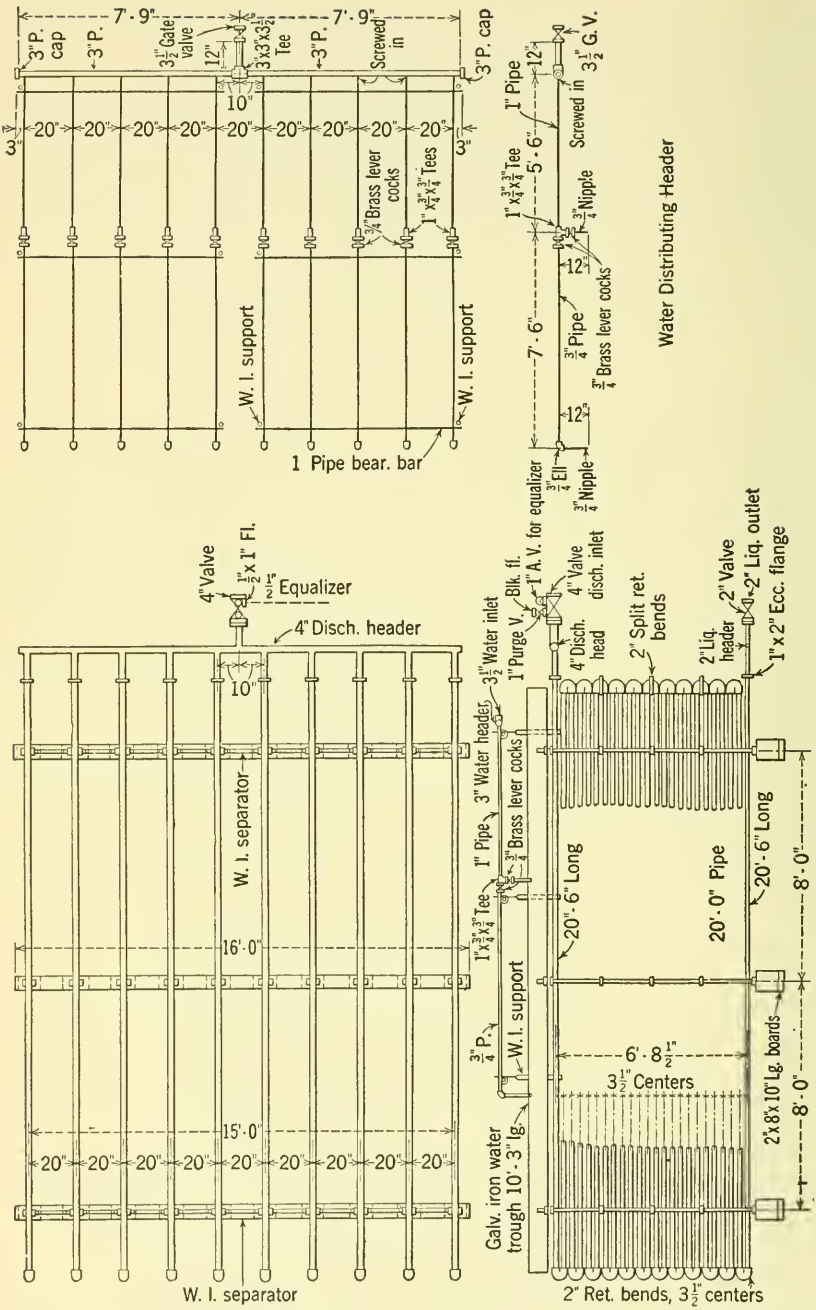



FIG. 228.—Condenser Piping—The Atmospheric Condenser.

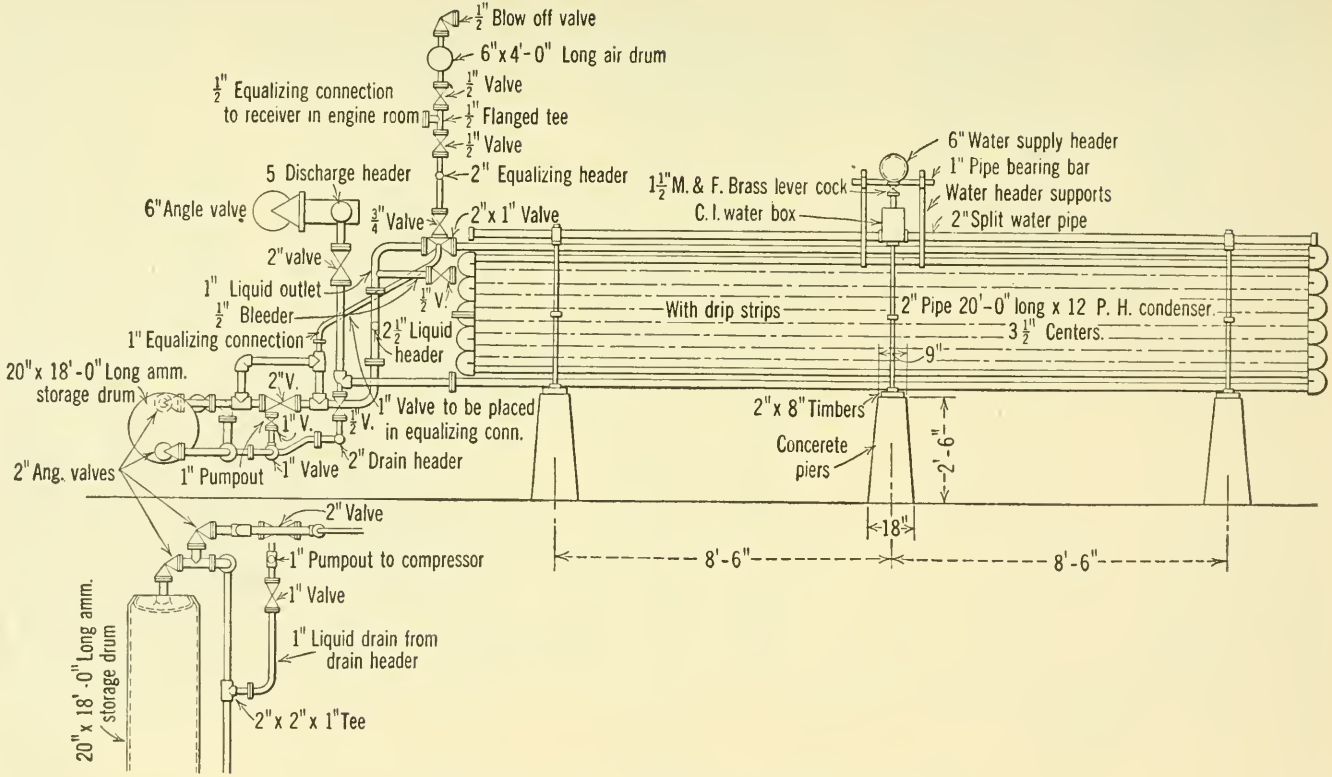



Fig. 229.—Condenser Piping—The Bleeder Condenser.

just one stand, or its equivalent, is used the piping is the simplest, but otherwise the discharge gas from the compressor enters a distributing header and the condensate flows through a collecting header to the liquid

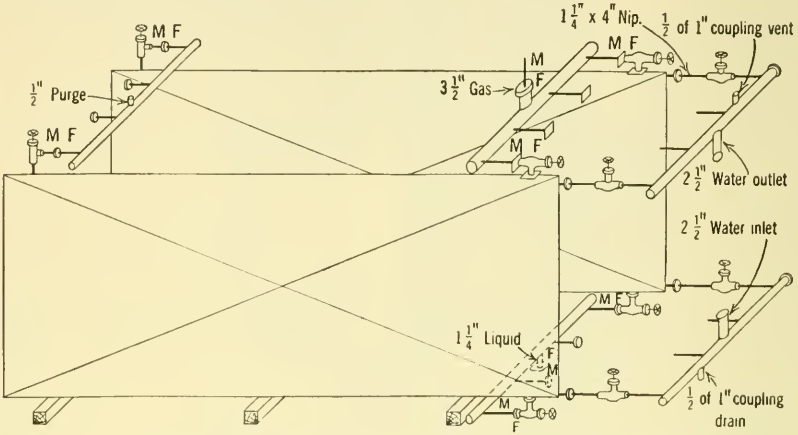



Fig. 230.—Condenser Piping—The Double Pipe Condenser.

receiver. Some manufacturers have a single stop valve on each header, as shown in Fig. 228, while others have stop valves on each stand to facilitate repairs on the condenser, as suggested in Fig. 230. The

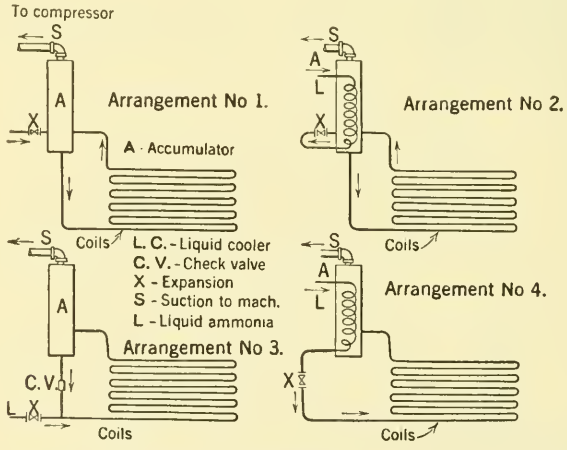



Fig. 231.—Four Piping Arrangements for the Flooded System.

condenser should be provided with connections for equalizing the pressure, with a purge valve, and especially in the large condensers, with a pump-out connection to the main compressor or to a special pump-out

compressor. The water supply connections are indicated in the figures also.

**The Liquid Line.**—The advisability of having a low temperature liquid refrigerant is shown elsewhere. Some plants are equipped with special liquid coolers, usually a double pipe arrangement for counter-flow and with the coldest water in the plant, but this is not common.

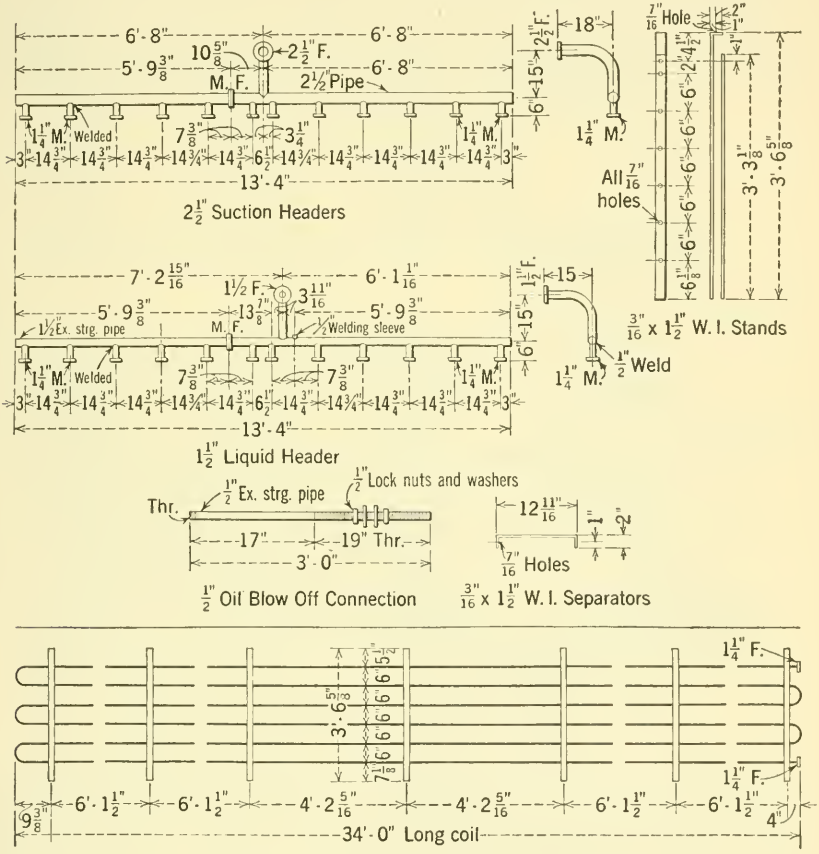



Fig. 232.—Piping and Headers for Can Ice Manufacture.

Neither is it common practice to insulate the liquid line, though this line is sometimes long and frequently placed in hot pipe shafts. There is no question but that insulation is advisable in many cases.

The size of the liquid pipe also depends on the length of the run. As a rule the velocity of flow of 3 to 6 ft. per sec. should be used. Short, self-contained plants can increase this value, and long lines, especially

where considerable vertical rise is required, should be designed using a minimum value. Too large a vertical rise should not be attempted, though on account of the lesser density of the liquid ammonia it will rise 50 per cent higher than will water with the same pressure. The loss of pressure at the expansion valve will be occasioned by pipe friction, velocity head and vertical lift.

**The Low Pressure Side.**—*Brine cooling.*—Whenever brine is cooled by the use of the shell and tube brine cooler for indirect refrigeration, the simplest pipe arrangement for the ammonia is provided (see Chapter VIII). It is self-contained and there is little danger of liquid return in

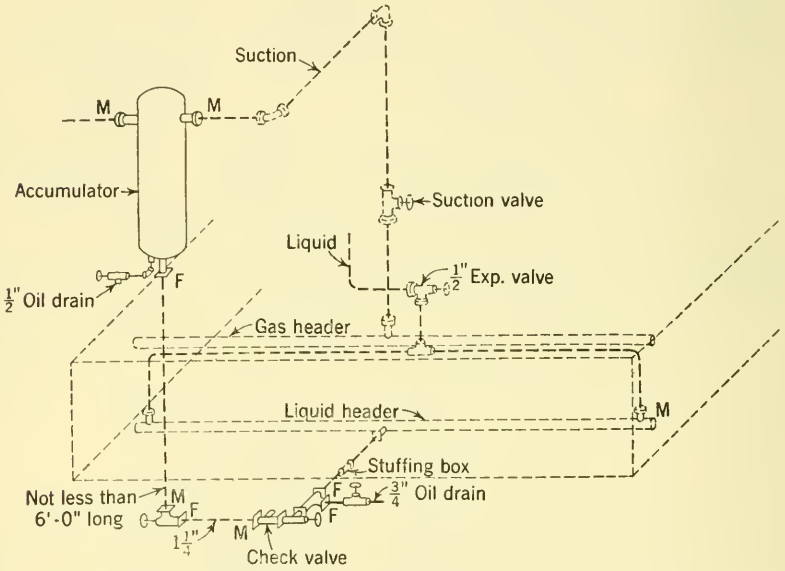



Fig. 233.—The Accumulator and Brine Tank Piping.

the suction pipe line to the compressor. In can ice making it is somewhat more complicated because of the use of the flooded system when direct expansion piping is used for the cooling of brine. Several piping arrangements for the flooded system are shown in Fig. 231. The flooded system embodies a device for precooling the liquid before it enters the expansion coils, and an accumulator to prevent liquid returning to the compressor thereby acting somewhat like the steam separator in steam engineering.

The brine itself is cooled by expansion piping, as shown by Fig. 232. In this case what is desired is a uniform temperature of the brine with a minimum obstruction to the cans. Because of the difficulty of getting

at the piping during operation it is well to use a welded pipe wherever practicable. Limits to welding lie in the ability to handle the coils unless the piping can be welded in place on the job. In can ice making plants using shell and tube brine coolers, the piping would be the same as in the indirect brine system.

**Cold Storage Piping.**—In cold storage piping the important factors are to secure the proper amount of piping with the least obstruction to

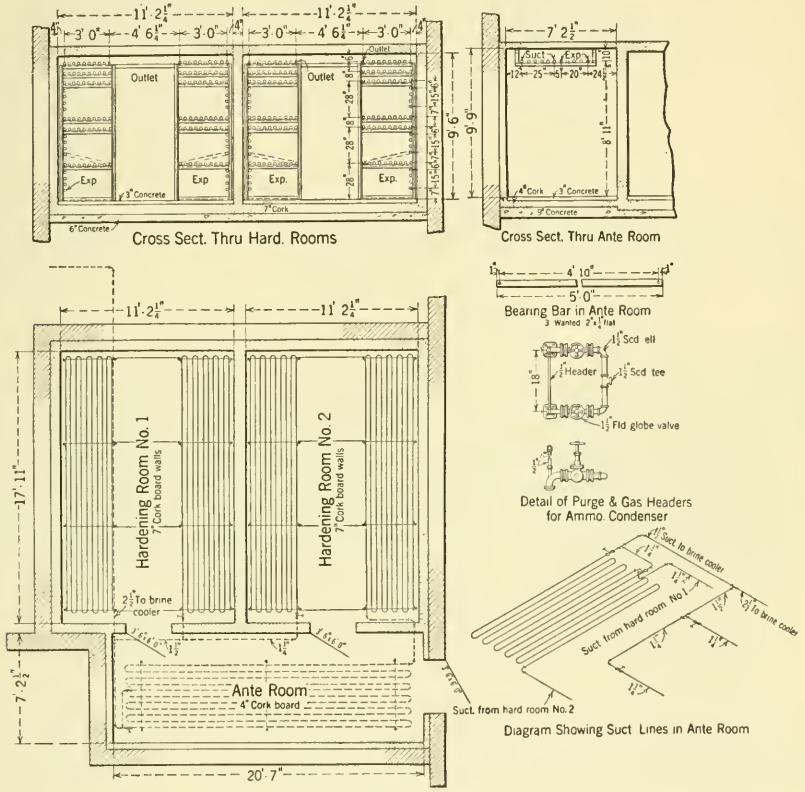

Cross Sect. Thru Hard. Rooms
Cross Sect. Thru Ante Room
Bearing Bar in Ante Room
Detail of Purge & Gas Headers for Ammo. Condenser
Diagram Showing Suct Lines in Ante Room
Hardening Room No. 1
Hardening Room No. 2
Ante Room
Outlet
Exp.
Suct. from hard room No. 2

FIG. 234.—Piping—The Sharp Freezer.

the storage of commodities and the least danger due to the melting of the frostation. Typical examples are shown in Figs. 234, 235 and 236 for small rooms and in Figs. 237 to 239 for larger rooms. It will be noticed that the larger rooms use overhead coils entirely. If the piping is not very heavy (in surface) it is usually arranged to be over aisles, but in any case galvanized iron drip pans should be placed under the coils with a suitable lead-off pipe to take the drips from the pipes should the

frostation melt off. How much pipe in series is an important matter to decide. The deciding factor is the ability of the gas to free itself and

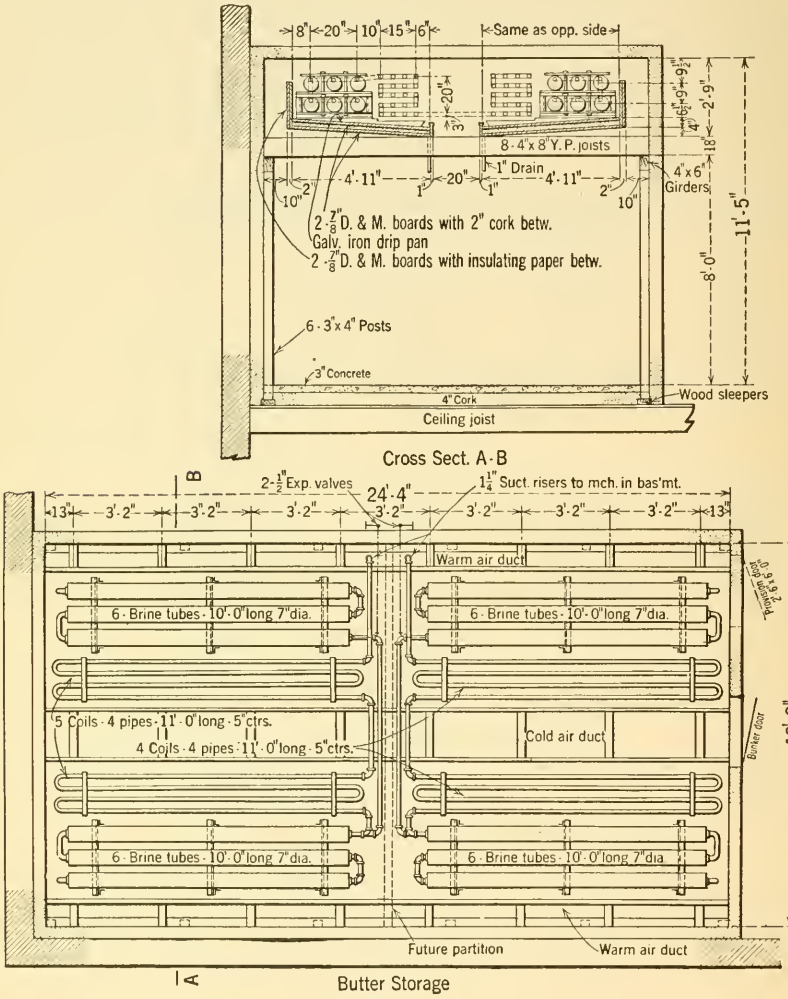
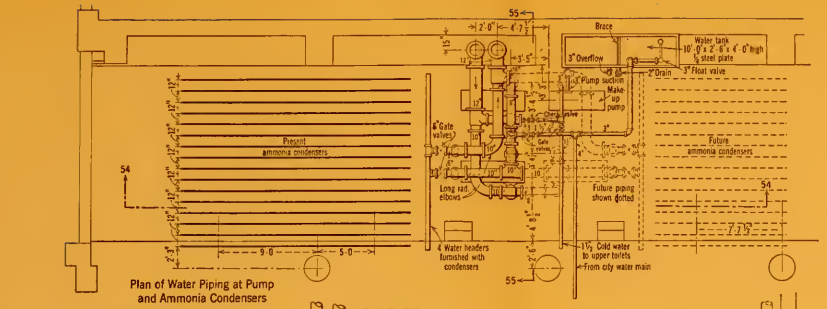

8" 20" 10" 15" 6" Same as opp. side
8 · 4 x 8" Y. P. joists
1" Drain
4" x 6" Girders
2 · 7/8" D. & M. boards with 2" cork betw.
Galv. iron drip pan
2 · 7/8" D. & M. boards with insulating paper betw.
6 · 3" x 4" Posts
3" Concrete
4" Cork
Wood sleepers
Ceiling joist
11'·5"
8'·0"
Cross Sect. A-B
2-1/2" Exp. valves
1 1/4" Suct. risers to mch. in bas'mt.
24'·4"
Warm air duct
6 · Brine tubes · 10'·0" long 7" dia.
5 Coils · 4 pipes · 11'·0" long · 5" ctrs.
4 Coils · 4 pipes · 11'·0" long · 5" ctrs.
Cold air duct
Bunker door
13'·8"
Future partition
Warm air duct
Butter Storage

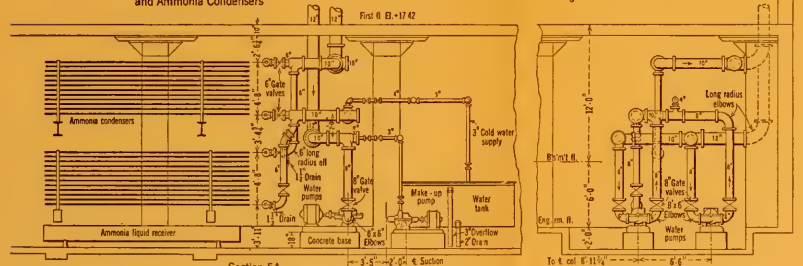
Fig. 235.—Piping—Butter Storage Using Brine Tubes.

return to the compressor without excessive pressure loss. The amount recommended is, as a maximum,

1100 ft. of 1-in. pipe,
1300 ft. of 1¼-in. pipe,
1600 ft. of 1½-in. pipe,
1900 ft. of 2-in. pipe.




Plan of Water Piping at Pump and Ammonia Condensers

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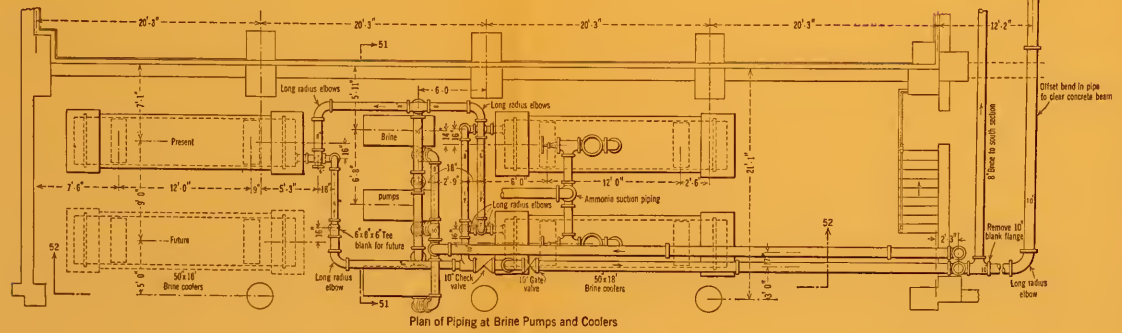
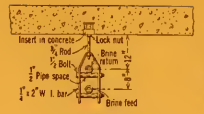
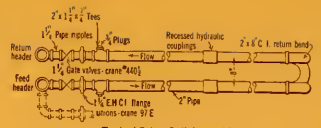


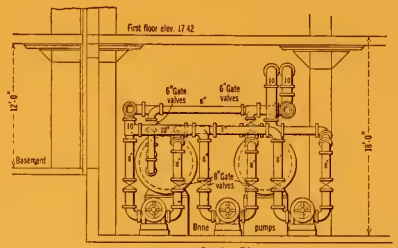

Plan of Piping at Brine Pumps and Coolers




Typical Header Hanger




Typical Brine Coil Assembly




Section 51

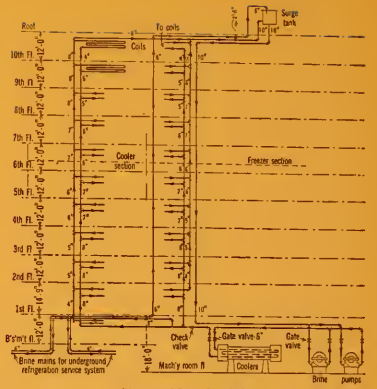
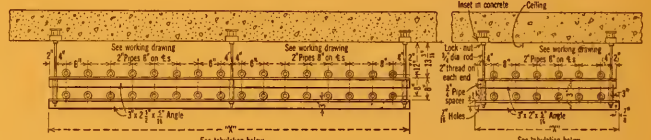
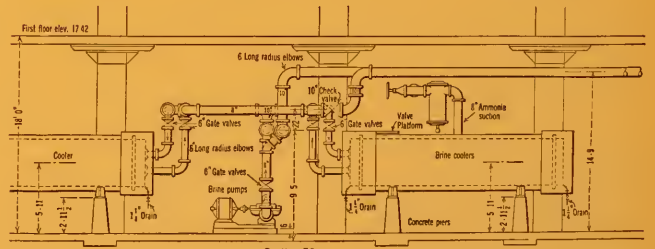



Diagram of Brine Circulation




Hangers "C" Hangers "A"




Section 52

Fig. 237.—Piping—Brine Details.

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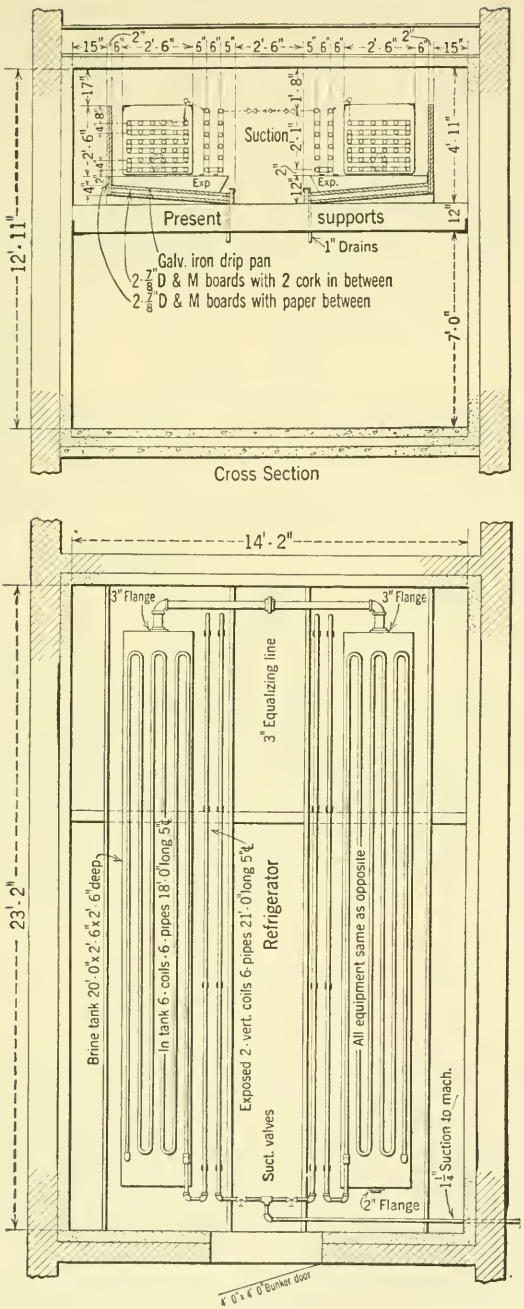



FIG. 236.—Piping—Direct Expansion and Brine Tank.

On the other hand, the pipe line must not be too short. The liquid fed into the pipe line must have an opportunity to boil out before it has to leave the evaporating coils. If it is not all evaporated some liquid will return to the compressor, an action that always causes operating trouble. With brine lines the case is different, as the brine is not subject to the bad effects from gas formation in the pipe. The length of pipe in series for brine varies with the operating conditions from 100 to 400 ft. The use of " top feed " and " bottom feed " for ammonia has been a disputed point for years. At present it is conceded by most engineers, if

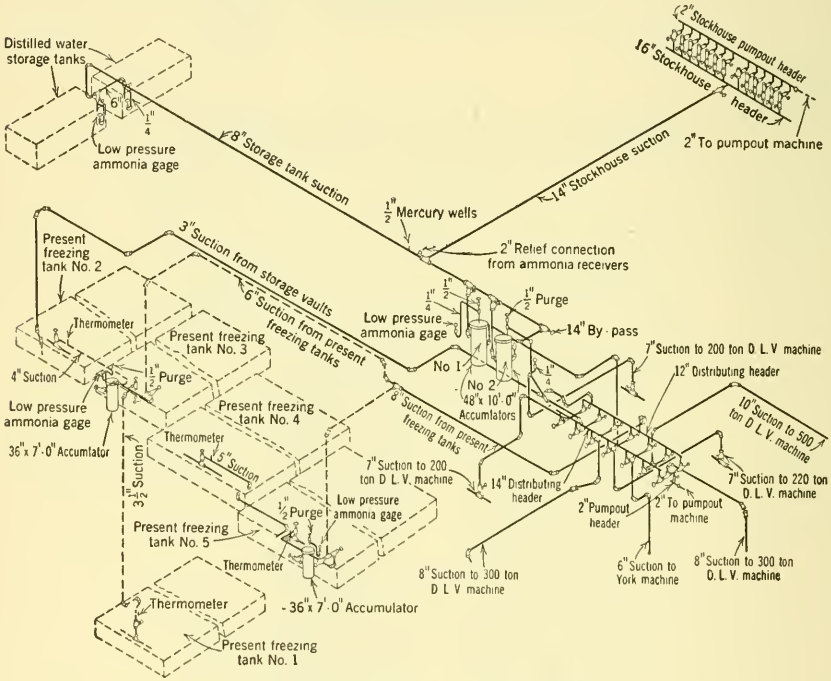
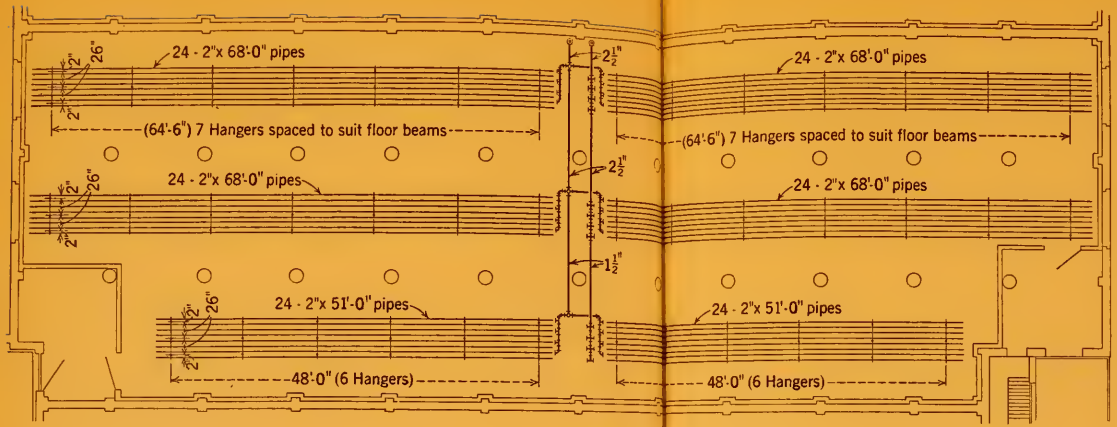
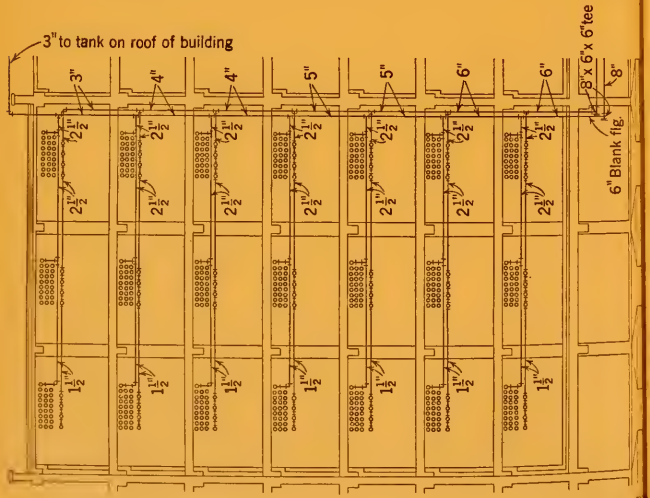

Distilled water storage tanks
Low pressure ammonia gage
8" Storage tank suction
2" Stockhouse pumpout header
16" Stockhouse header
14" Stockhouse suction
2" To pumpout machine
½" Mercury wells
2" Relief connection from ammonia receivers
Present freezing tank No. 2
3" Suction from storage vaults
6" Suction from present freezing tanks
Thermometer
Present freezing tank No. 3
4" Suction
½" Purge
Low pressure ammonia gage
36" x 7'-0" Accumulator
Present freezing tank No. 4
Thermometer
3½" Suction
½" Suction
½" Purge
Low pressure ammonia gage
Present freezing tank No. 5
Thermometer
Thermometer
36" x 7'-0" Accumulator
Present freezing tank No. 1
Low pressure ammonia gage
½" Purge
14" By-pass
No 1
No 2
48" x 10'-0" Accumulators
8" Suction from present freezing tanks
7" Suction to 200 ton D L V. machine
14" Distributing header
8" Suction to 300 ton D L V machine
7" Suction to 500 ton D L V. machine
12" Distributing header
7" Suction to 220 ton D. L. V. machine
2" Pumpout header
2" To pumpout machine
6" Suction to York machine
8" Suction to 300 ton D. L. V. machine

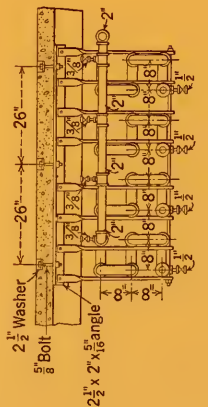
Fig. 239.—Piping—Ammonia Mains.

a suitable protection against liquid returning to the compressor is installed, that bottom feed gives better heat transfer and is the better method of operation. The arrangement of the piping should be such as would permit control of the temperature by increasing or decreasing the amount of piping to suit. This can be done by the arrangement of the coils so that more or less can be turned into action and enough must be supplied to carry the full load.

The pipes are supported in a number of ways. For cold storage work the usual manner is to suspend them from the ceiling. In the case of


24 - 2"x 68'-0" pipes
(64'-6") 7 Hangers spaced to suit floor beams
24 - 2"x 68'-0" pipes
24 - 2"x 51'-0" pipes
48'-0" (6 Hangers)
24 - 2"x 68'-0" pipes
(64'-6") 7 Hangers spaced to suit floor beams
24 - 2"x 68'-0" pipes
24 - 2"x 51'-0" pipes
48'-0" (6 Hangers)



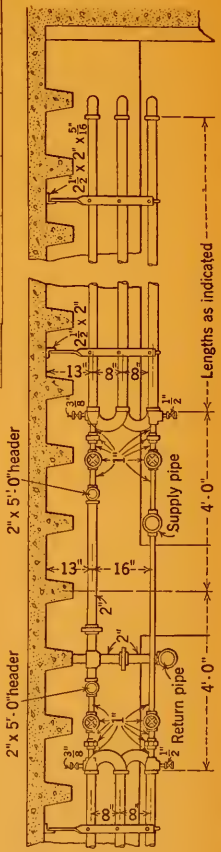




FIG. 238.—Brine Details.

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pipe decks, suitable stands from the decks can be used, and in some special cases the piping may be supported from the walls and from the floor. Examples of piping supports are given in the figures.

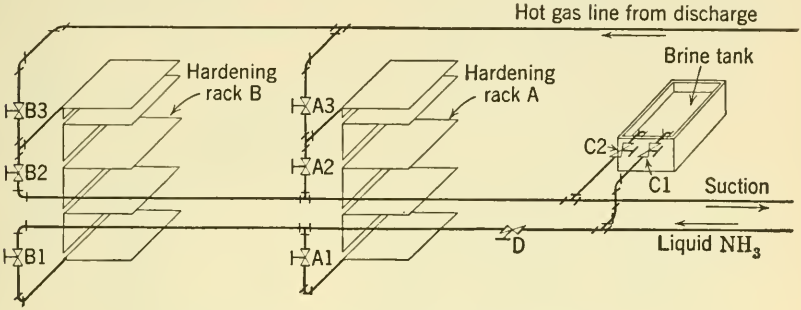



Fig. 240.—Piping—Ice Cream Hardening.

The *ice cream hardening* room is a problem by itself. As usually designed it is very heavily piped (Fig. 240 and Chapter XIV), and the piping is arranged so as to make shelves on which the ice cream is placed in bricks, fancy shapes and in bulk. It is very nearly always a direct

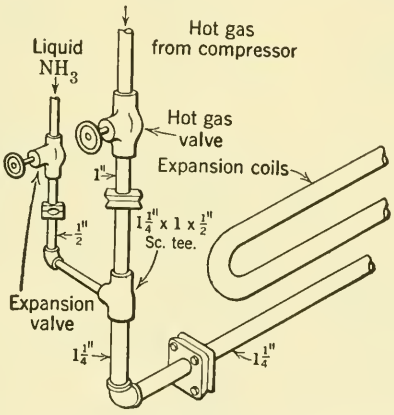



Fig. 241.—Piping—Defrosting Large Installations.

expansion job with the expansion valve arranged for bottom feed of the liquid. The sharp freezer is a term used for a low temperature room. It also is very heavily piped,[1] and may or may not have the piping arranged in the form of shelves. A fish freezer is likely to have shelves so that the fish may be placed in pans on the piping.

**Defrosting.**—Defrosting of the evaporating coils can be accomplished in a number of ways. The frost may be scraped off from time

[1] See Chapter XIV for the usual values for the amount of piping to be used.

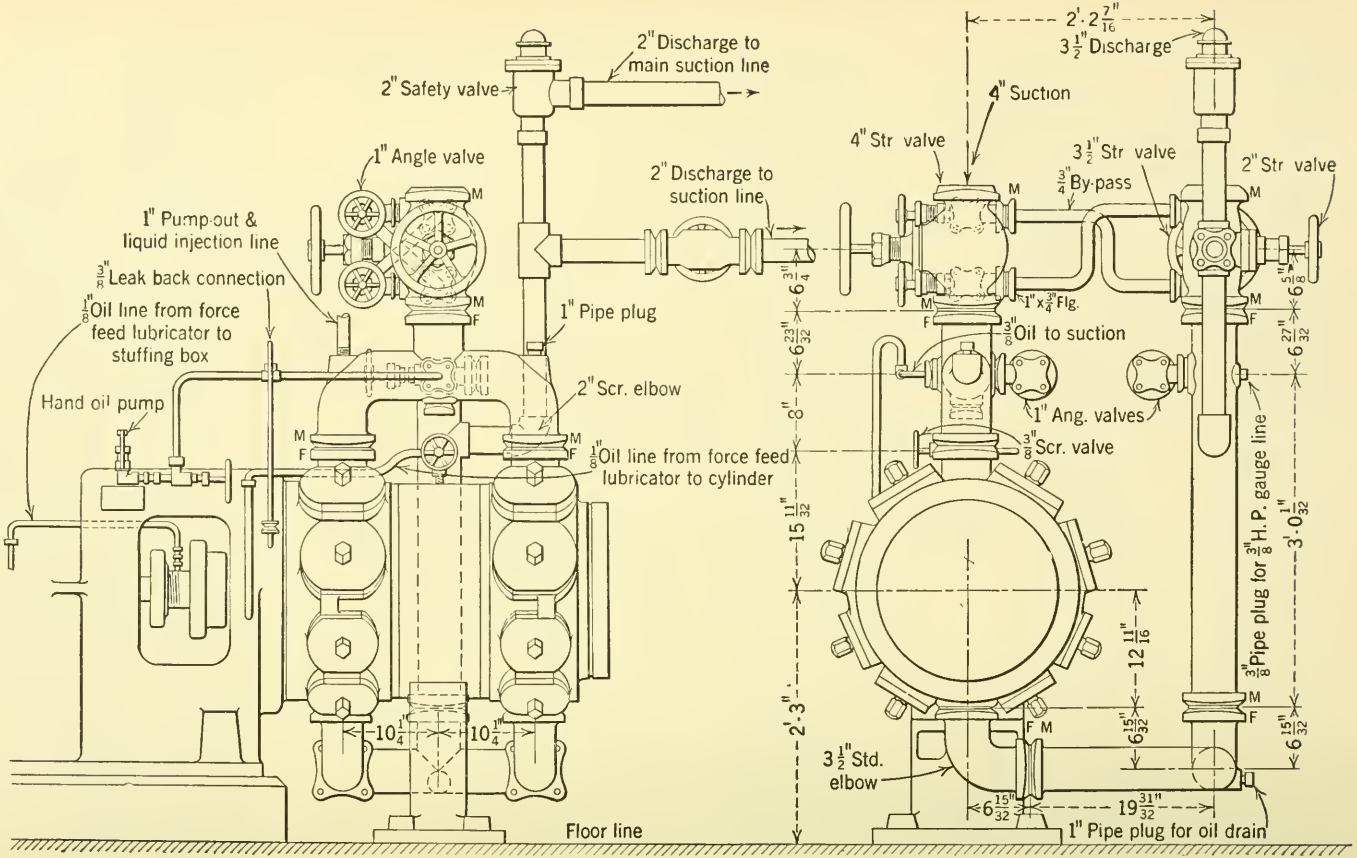



FIG. 242.—Piping—The Horizontal Double Acting Compressor.

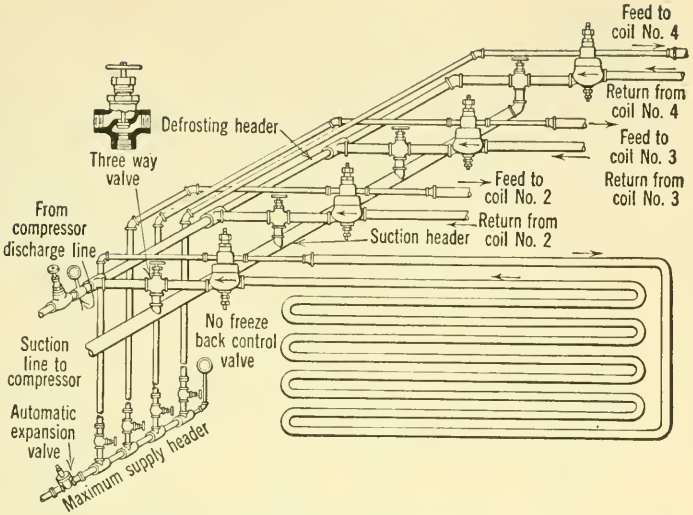

Feed to coil No. 4
Return from coil No. 4
Feed to coil No. 3
Return from coil No. 3
Feed to coil No. 2
Return from coil No. 2
Suction header
Defrosting header
Three way valve
From compressor discharge line
No freeze back control valve
Suction line to compressor
Automatic expansion valve
Maximum supply header

Fig. 243.—Piping—The Non-Freeze Back.

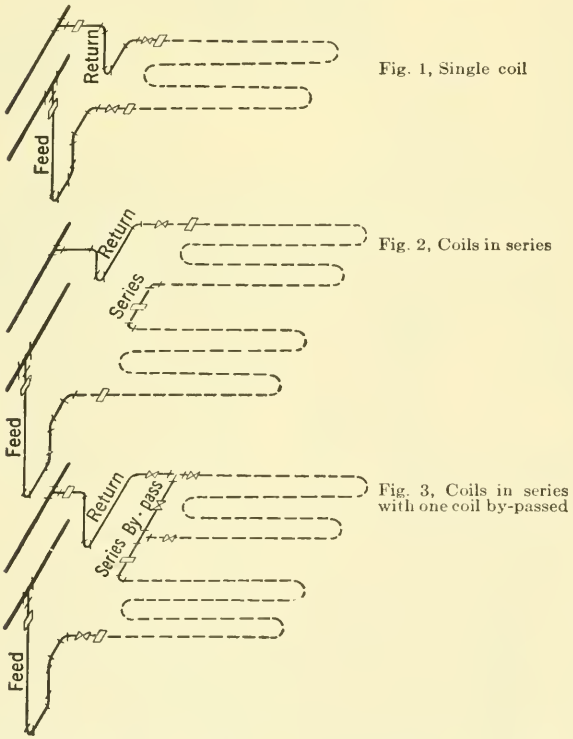



Fig. 244.—Piping Connections to Direct Expansion Coils.

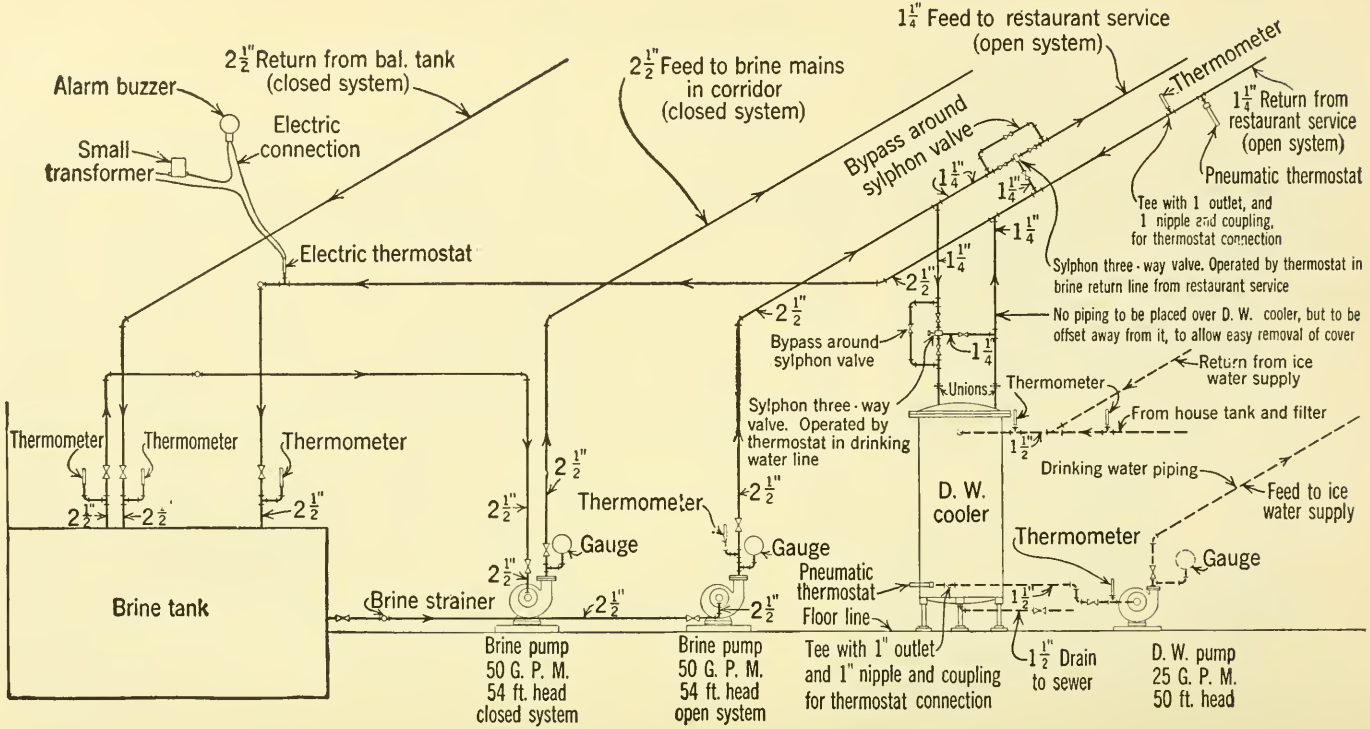

Alarm buzzer
Small transformer
2½" Return from bal. tank (closed system)
Electric connection
Electric thermostat
2½" Feed to brine mains in corridor (closed system)
1¼" Feed to restaurant service (open system)
Bypass around sylphon valve
Thermometer
1¼" Return from restaurant service (open system)
Pneumatic thermostat
Tee with 1 outlet, and 1 nipple and coupling, for thermostat connection
Sylphon three-way valve. Operated by thermostat in brine return line from restaurant service
No piping to be placed over D. W. cooler, but to be offset away from it, to allow easy removal of cover
Bypass around sylphon valve
Unions
Thermometer
Return from ice water supply
From house tank and filter
Sylphon three-way valve. Operated by thermostat in drinking water line
Drinking water piping
Feed to ice water supply
D. W. cooler
Thermometer
Gauge
Pneumatic thermostat
Floor line
Thermometer
Thermometer
Thermometer
Gauge
Gauge
Brine strainer
Brine tank
Brine pump 50 G. P. M. 54 ft. head closed system
Brine pump 50 G. P. M. 54 ft. head open system
Tee with 1" outlet and 1" nipple and coupling for thermostat connection
1½" Drain to sewer
D. W. pump 25 G. P. M. 50 ft. head

FIG. 245.—Piping—The Automatic Brine System.

to time, and occasionally brine may be arranged to trickle over the pipes. By reversing the connections on the compressor, hot gas can be pumped into the low pressure coils (Fig. 234) or a separate defrosting, and pump-out line can be installed.

Piping for defrosting is shown in Figs. 240, 241 and 243. These are arrangements for permitting hot gas from the compressor to enter the frosted piping.

**Non-freeze Back.**—A control to prevent automatically the return

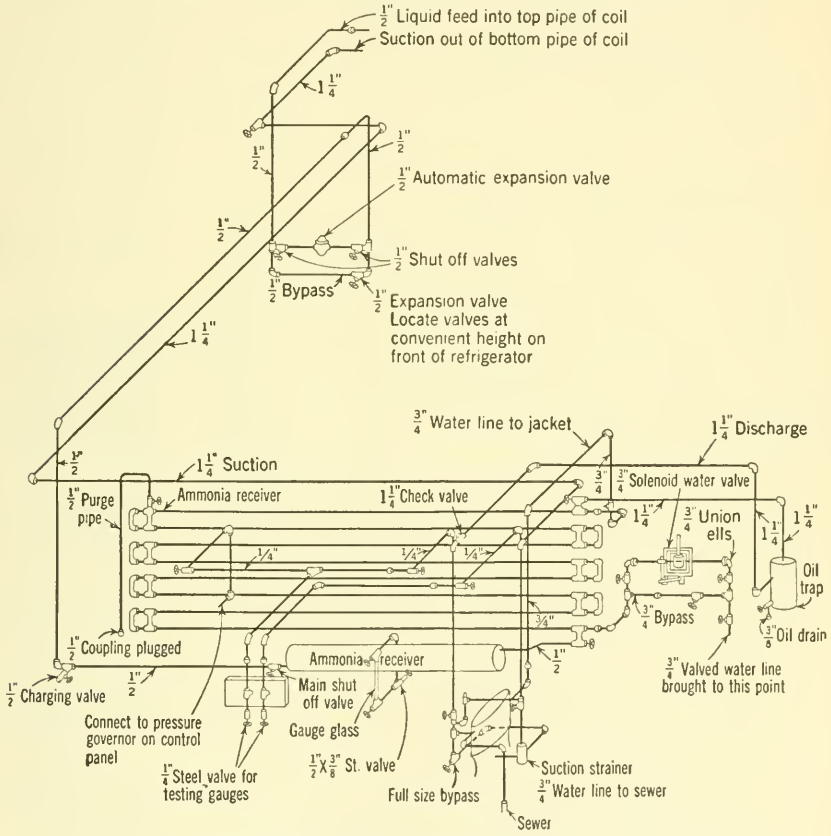



Fig. 246.—Piping—The Automatic Direct Expansion System.

of liquid to the compressor can be accomplished by a control of the temperature in the suction return line. This is done by the device shown in Fig. 236, or the same result can be obtained by means of an automatic control, by winding the bulb of the thermostat around the suction pipe and by setting the contacts for the temperature required in conjunction with a device for controlling the liquid feed.

## TABLE 85

### Dimensions of Standard Steel Pipe

| Diameter, Inches | | | Thickness, Inches | Circumference, Inches | | Transverse Areas, Square Inches | | | Length of Pipe per Square Foot | | Length of Pipe Containing One Cubic Foot, Feet | Weight per Foot of Length, Pounds | Number of Threads per Inch of Screw | Contents per Foot of Length, Gallons |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Nominal internal | Actual external | Actual internal | | External | Internal | External | Internal | Metal | External surface, feet | Internal surface, feet | | | | |
| ⅛ | .405 | .269 | .068 | 1.272 | .848 | .129 | .0573 | .0717 | 9.44 | 14.15 | 2513.0 | .244 | 27 | .0006 |
| ¼ | .54 | .364 | .088 | 1.696 | 1.144 | .229 | .1041 | .1249 | 7.075 | 10.49 | 1383.3 | .424 | 18 | .0026 |
| ⅜ | .675 | .493 | .091 | 2.121 | 1.552 | .358 | .1917 | .1663 | 5.657 | 7.73 | 751.2 | .567 | 18 | .0057 |
| ½ | .84 | .622 | .109 | 2.639 | 1.957 | .554 | .3048 | .2492 | 4.547 | 6.13 | 472.4 | .850 | 14 | .0102 |
| ¾ | 1.05 | .824 | .113 | 3.299 | 2.589 | .866 | .5333 | .3327 | 3.637 | 4.635 | 270.0 | 1.130 | 14 | .0230 |
| 1 | 1.315 | 1.049 | .133 | 4.131 | 3.292 | 1.358 | .8626 | .4954 | 2.904 | 3.645 | 166.9 | 1.678 | 11½ | .0408 |
| 1¼ | 1.66 | 1.38 | .14 | 5.215 | 4.335 | 2.164 | 1.496 | .668 | 2.301 | 2.768 | 96.25 | 2.272 | 11½ | .0638 |
| 1½ | 1.9 | 1.16 | .145 | 5.969 | 5.061 | 2.835 | 2.038 | .797 | 2.01 | 2.371 | 70.66 | 2.717 | 11½ | .0918 |
| 2 | 2.375 | 2.067 | .154 | 7.461 | 6.494 | 4.43 | 3.356 | 1.074 | 1.608 | 1.848 | 42.91 | 3.652 | 11½ | .1632 |
| 2½ | 2.875 | 2.469 | .203 | 9.032 | 7.753 | 6.492 | 4.784 | 1.708 | 1.328 | 1.547 | 30.1 | 5.793 | 8 | .2550 |
| 3 | 3.5 | 3.068 | .216 | 10.996 | 9.636 | 9.621 | 7.388 | 2.243 | 1.091 | 1.245 | 19.5 | 7.575 | 8 | .3673 |
| 3½ | 4.0 | 3.548 | .226 | 12.556 | 11.146 | 12.566 | 9.887 | 2.679 | .955 | 1.077 | 14.57 | 9.109 | 8 | .4998 |
| 4 | 4.5 | 4.026 | .237 | 14.137 | 12.648 | 15.904 | 12.73 | 3.174 | .849 | .949 | 11.31 | 10.790 | 8 | .6528 |
| 4½ | 5.0 | 4.506 | .247 | 15.708 | 14.162 | 19.635 | 15.961 | 3.674 | .764 | .848 | 9.02 | 12.538 | 8 | .8263 |
| 5 | 5.563 | 5.047 | .258 | 17.477 | 15.849 | 24.306 | 19.99 | 4.316 | .687 | .757 | 7.2 | 14.617 | 8 | 1.020 |
| 6 | 6.625 | 6.065 | .28 | 20.813 | 19.054 | 34.472 | 28.888 | 5.584 | .577 | .63 | 4.98 | 18.974 | 8 | 1.469 |
| 7 | 7.625 | 7.023 | .301 | 23.955 | 22.063 | 45.664 | 38.738 | 6.926 | .501 | .544 | 3.72 | 28.654 | 8 | 1.999 |
| 8 | 8.625 | 7.981 | .322 | 27.096 | 25.076 | 58.426 | 50.04 | 8.386 | .443 | .478 | 2.88 | 28.554 | 8 | 2.611 |
| 9 | 9.625 | 8.941 | .342 | 30.238 | 28.076 | 72.76 | 62.73 | 10.03 | .397 | .427 | 2.29 | 33.907 | 8 | 3.300 |
| 10 | 10.75 | 10.020 | .365 | 33.772 | 31.477 | 90.763 | 78.839 | 11.924 | .355 | .382 | 1.82 | 40.483 | 8 | 4.081 |
| 11 | 11.75 | 11.0 | .375 | 37.699 | 35.343 | 113.098 | 99.402 | 13.696 | .318 | .339 | 1.456 | 45.557 | 8 | |
| 12 | 12.75 | 12.0 | .375 | 40.055 | 37.7 | 127.677 | 113.098 | 14.579 | .299 | .319 | 1.27 | 49.562 | 8 | |

| | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 13 | 14.0 | 13.25 | .375 | 43.982 | 41.626 | 153.938 | 137.88 | 16.051 | .273 | .288 | 1.04 | 54.568 | 8 | |
| 14 | 15.0 | 14.25 | .375 | 47.124 | 44.768 | 176.715 | 159.485 | 17.23 | .255 | .268 | .903 | 58.573 | 8 | |
| 15 | 16.0 | 15.25 | .375 | 50.265 | 47.909 | 201.062 | 182.655 | 18.407 | .239 | .250 | .788 | 62.579 | 8 | |
| 17 | 18.0 | 17.25 | .375 | 56.549 | 54.192 | 254.47 | 233.706 | 20.764 | .212 | .221 | .616 | 70.589 | | |
| 19 | 20.0 | 19.25 | .375 | 62.832 | 60.476 | 314.16 | 291.04 | 23.12 | .191 | .198 | .495 | 78.599 | | |
| 21 | 22.0 | 21.25 | .375 | 69.115 | 66.759 | 380.134 | 354.657 | 25.477 | .174 | .179 | .406 | 86.609 | | |
| 23 | 24.0 | 23.25 | .375 | 75.398 | 73.042 | 452.39 | 424.558 | 27.832 | .159 | .164 | .339 | 94.619 | | |

## Dimensions of Extra-heavy Steel Pipe

| Diameter, Inches | | | Thickness, Inches | Nearest Wire Gage, Number | Circumference, Inches | | Transverse Areas, Square Inches | | | Length of Pipe, per Square Foot | | Nominal Weight per Foct, Pounds |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Nominal internal | Actual external | Actual internal | | | External | Internal | External | Internal | Metal | External surface, feet | Internal surface, feet | |
| ⅛ | .405 | .215 | .095 | 12½ | 1.272 | .644 | .129 | .033 | .086 | 9.433 | 18.632 | .314 |
| ¼ | .54 | .302 | .119 | 11 | 1.696 | .924 | .229 | .068 | .161 | 7.075 | 12.986 | .535 |
| ⅜ | .675 | .423 | .126 | 10½ | 2.121 | 1.323 | .358 | .139 | .219 | 5.657 | 9.07 | .738 |
| ½ | .84 | .546 | .147 | 9 | 2.639 | 1.703 | .554 | .231 | .323 | 4.547 | 7.046 | 1.087 |
| ¾ | 1.05 | .742 | .154 | 8½ | 3.299 | 2.312 | .866 | .4324 | .414 | 3.637 | 5.109 | 1.473 |
| 1 | 1.315 | .957 | .179 | 7 | 4.131 | 2.988 | 1.358 | .71 | .648 | 2.904 | 4.016 | 2.171 |
| 1¼ | 1.66 | 1.278 | .191 | 6½ | 5.215 | 3.996 | 2.164 | 1.271 | .893 | 2.301 | 3.003 | 2.996 |
| 1½ | 1.9 | 1.500 | .200 | 6 | 5.969 | 4.694 | 2.835 | 1.753 | 1.082 | 2.01 | 2.556 | 3.631 |
| 2 | 2.375 | 1.939 | .218 | 5 | 7.461 | 6.073 | 4.43 | 2.935 | 1.495 | 1.608 | 1.975 | 5.022 |
| 2½ | 2.875 | 2.323 | .276 | 2 | 9.032 | 7.273 | 6.492 | 4.209 | 2.283 | 1.328 | 1.649 | 7.661 |
| 3 | 3.5 | 2.900 | .300 | 1 | 10.996 | 9.085 | 9.621 | 6.569 | 3.052 | 1.091 | 1.328 | 10.252 |
| 3½ | 4.0 | 3.364 | .318 | 0 | 12.566 | 10.549 | 12.566 | 8.856 | 3.71 | .955 | 1.137 | 12.505 |
| 4 | 4.5 | 3.826 | .337 | 0 | 14.137 | 11.995 | 15.904 | 11.449 | 4.455 | .849 | 1.0 | 14.983 |
| 5 | 5.563 | 4.813 | .375 | 00 | 17.477 | 15.120 | 24.306 | 18.193 | 6.12 | .687 | .793 | 20.778 |
| 6 | 6.625 | 5.761 | .432 | 000 | 20.813 | 18.064 | 34.472 | 25.967 | 8.505 | .577 | .664 | 28.573 |

**The Radiator Section.**—The radiator section, made of a thin walled cast iron or semi-steel, can be manufactured to hold ammonia, and it has become popular in certain sections of the United States for certain kinds of refrigeration. For ammonia the section is made with the tongue-and-groove flanged joint, and can be erected in the usual manner (as for heating installations) with the columns vertical, or, for shelving in ice cream hardening rooms, with the columns horizontal. Tests[2] indicate that a section of this nature will withstand a heavy hydraulic

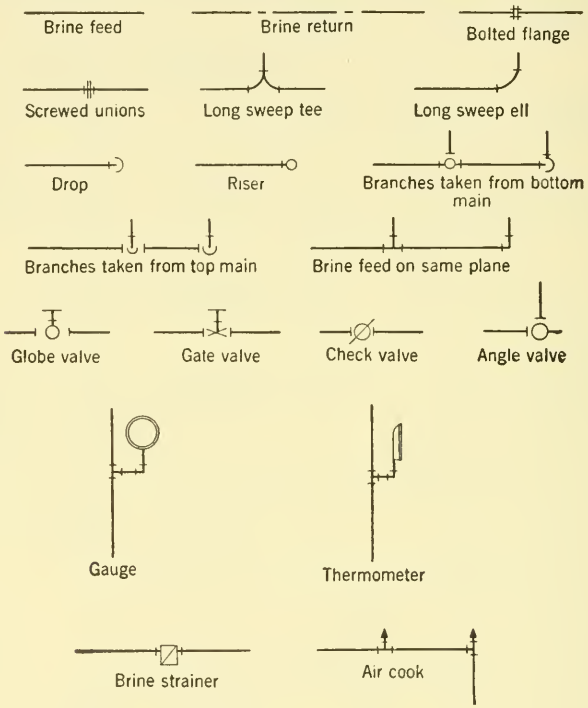



Fig. 247.—Conventional Piping Symbols.

pressure and will carry a heavy load on a long span without leaks or without showing signs of failure. The advantages of cast sections are in the ease of erection, the probable better heat transfer[3] and the greater ease with which the section frees itself of gas as compared with common piping. The application of cast iron or semi-steel sections to ammonia refrigeration is similar to the arrangement used in steam heating.

[2] The Flexibility of Cast Iron Sections for Ammonia, H. J. Macintire, Journal of the American Society of Refrigerating Engineers, 1922.

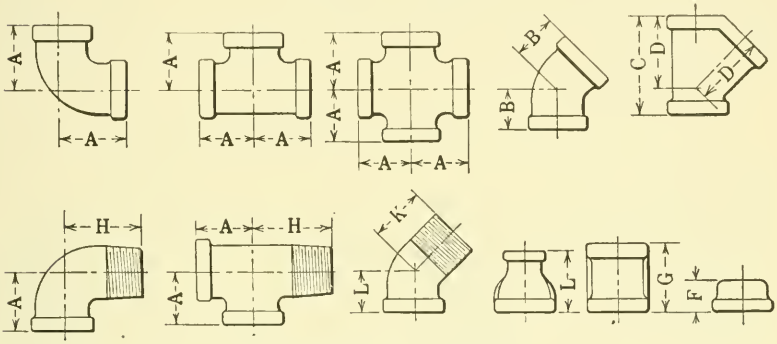
[3] Heat Transfer in Cast Iron Sections, H. J. Macintire, Refrigerating Engineering, Dec., 1924.

Figures 245 and 246 show typical piping for automatic operation of brine and ammonia systems. Table 85 gives the dimensions and cross-sectional areas of full weight and extra heavy wrought iron and steel pipe, and Table 86 gives the general dimensions of standard malleable iron screw fittings. Figures 247 gives typical conventional symbols for piping.

TABLE 86

STANDARD MALLEABLE IRON-SCREWED FITTINGS

General Dimensions




All Dimensions Given in Inches

| Size | $\frac{1}{8}$ | $\frac{1}{4}$ | $\frac{3}{8}$ | $\frac{1}{2}$ | $\frac{3}{4}$ | 1 | $1\frac{1}{4}$ | $1\frac{1}{2}$ | 2 | $2\frac{1}{2}$ | 3 | $3\frac{1}{2}$ | 4 | $4\frac{1}{2}$ | 5 | 6 | 7 | 8 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| *A* | $\frac{11}{16}$ | $\frac{13}{16}$ | $\frac{15}{16}$ | $1\frac{1}{8}$ | $1\frac{5}{16}$ | $1\frac{7}{16}$ | $1\frac{3}{4}$ | $1\frac{15}{16}$ | $2\frac{1}{4}$ | $2\frac{11}{16}$ | $3\frac{1}{8}$ | $3\frac{7}{16}$ | $3\frac{3}{4}$ | $4\frac{1}{16}$ | $4\frac{7}{16}$ | $5\frac{1}{8}$ | $5\frac{13}{16}$ | $6\frac{1}{2}$ |
| *B* | ... | $\frac{3}{4}$ | $\frac{13}{16}$ | $\frac{7}{8}$ | 1 | $1\frac{1}{8}$ | $1\frac{5}{16}$ | $1\frac{7}{16}$ | $1\frac{11}{16}$ | $1\frac{15}{16}$ | $2\frac{3}{16}$ | $2\frac{3}{8}$ | $2\frac{5}{8}$ | $2\frac{13}{16}$ | $3\frac{1}{16}$ | $3\frac{7}{16}$ | $3\frac{7}{8}$ | $4\frac{1}{4}$ |
| *C* | ... | ... | ... | $2\frac{1}{2}$ | $2\frac{7}{8}$ | $3\frac{7}{16}$ | $4\frac{1}{16}$ | $4\frac{1}{2}$ | $5\frac{7}{16}$ | $6\frac{1}{4}$ | $7\frac{1}{4}$ | ... | $8\frac{7}{8}$ | | | | | |
| *D* | ... | ... | ... | $1\frac{11}{16}$ | 2 | $2\frac{7}{16}$ | $2\frac{15}{16}$ | $3\frac{5}{16}$ | $4\frac{1}{16}$ | $4\frac{11}{16}$ | $5\frac{9}{16}$ | ... | $6\frac{15}{16}$ | | | | | |
| *E* | ... | 1 | $1\frac{1}{8}$ | $1\frac{1}{4}$ | $1\frac{7}{16}$ | $1\frac{11}{16}$ | $2\frac{1}{16}$ | $2\frac{5}{16}$ | $2\frac{13}{16}$ | $3\frac{1}{4}$ | $3\frac{11}{16}$ | 4 | $4\frac{3}{8}$ | | | | | |
| *F* | ... | $\frac{5}{8}$ | $\frac{3}{4}$ | $\frac{7}{8}$ | $1\frac{1}{16}$ | $1\frac{3}{16}$ | $1\frac{1}{4}$ | $1\frac{5}{16}$ | $1\frac{7}{16}$ | $1\frac{5}{8}$ | $1\frac{3}{4}$ | $1\frac{15}{16}$ | 2 | ... | $2\frac{5}{16}$ | $2\frac{9}{16}$ | | |
| *G* | ... | $1\frac{1}{16}$ | $1\frac{3}{16}$ | $1\frac{5}{16}$ | $1\frac{1}{2}$ | $1\frac{11}{16}$ | $1\frac{15}{16}$ | $2\frac{1}{8}$ | $2\frac{1}{2}$ | $2\frac{7}{8}$ | $3\frac{3}{16}$ | | | | | | | |
| *H* | $1\frac{1}{8}$ | $1\frac{5}{16}$ | $1\frac{7}{16}$ | $1\frac{5}{8}$ | $1\frac{7}{8}$ | $2\frac{1}{8}$ | $2\frac{1}{2}$ | $2\frac{11}{16}$ | $3\frac{3}{16}$ | $3\frac{13}{16}$ | $4\frac{1}{2}$ | ... | $5\frac{11}{16}$ | | | | | |
| *K* | ... | $\frac{15}{16}$ | $1\frac{1}{16}$ | $1\frac{3}{16}$ | $1\frac{5}{16}$ | $1\frac{1}{2}$ | $1\frac{11}{16}$ | $1\frac{7}{8}$ | $2\frac{1}{4}$ | ... | 3 | ... | $3\frac{3}{4}$ | | | | | |
| *L* | ... | $\frac{11}{16}$ | $\frac{11}{16}$ | $\frac{13}{16}$ | $\frac{15}{16}$ | $1\frac{1}{16}$ | $1\frac{1}{4}$ | $1\frac{3}{8}$ | $1\frac{11}{16}$ | ... | $2\frac{1}{8}$ | | | | | | | |

Do not use for reducing fittings

## CHAPTER XIII

## ICE MAKING

**Natural Ice.**—Natural ice is less used every year. Where a few years ago it was the chief refrigerating agent it has now become almost negligible, as can be seen from the table (Table 87) which gives the amount of natural ice cut and stored for the last twenty years. The reasons for this falling off in the cutting of natural ice are several. The first and most important reason, perhaps, is its unreliability. Ice is a necessity, and at times the natural ice crop fails because of an open winter. Under these circumstances the hold-over from the previous winter hardly supplies the demand. Besides this, the cost of transportation is now much greater than heretofore and ice is a bulky commodity. Thus, although the railroads still transport a heavy tonnage, the average distance is not great. A lesser reason, but of some force even then, for the decline in the use of natural ice, is the fear of a contaminated source for the ice. This is, however, not a matter of serious concern as it has been well determined that freezing frees the ice of impurities, and when it is not used for several months the bacteria are mostly killed before the ice is consumed. At present natural ice is used chiefly for car and boat icing, where the appearance of the ice is not a factor.

**Manufactured Ice.**—The artificial ice industry, including plants, buildings and delivery equipment, is estimated (1924) to comprise nearly a billion dollars' investment. It is placed ninth in point of industrial investment in the United States, and there is reason to believe that the growth is going to continue, as more organized efforts are made to increase ice sales, especially to the household trade.[1]

In 1922 there were listed (Ice and Refrigeration Blue Book) 5500 plants manufacturing ice, with a daily total capacity of 223,000 tons. In this compilation 55.9 per cent was distilled water ice, 41.4 per cent was raw water can ice, and 2.7 per cent was plate ice, and of the plants engaged in the production of ice primarily 85 per cent had compression

[1] The effect of the household refrigerating machine is beginning (1927) to be felt in the larger cities, and in some localities the electric driven two, one and the fractional tonnage compressors have practically replaced ice in the wholesale trade for soda fountain, groceries and markets.

## TABLE 87

### Recapitulation

| | Capacity of Tons | Tons Harvested |
|---|---|---|
| Total Hudson River Houses | 1,823,321 | 238,083 |
| Total Lakes and Ponds | 194,804 | 4,548 |
| | 2,018,125 | 242,631 |

### Record of Previous Years

| Year, Totals | Capacity in Tons | Tons Harvested |
|---|---|---|
| 1923–1924 | 2,018,125 | 242,631 |
| 1922–1923 | 2,320,064 | 642,851* |
| 1921–1922 | 2,508,798 | 1,029,808 |
| 1920–1921 | 2,698,032 | 689,993† |
| 1919–1920 | 2,800,939 | 1,237,409 |
| 1918–1919 | 2,921,927 | 690,353 |
| 1917–1918 | 3,066,703 | 2,537,482 |
| 1916–1917 | 3,043,127 | 2,001,150 |
| 1915–1916 | 3,201,943 | 1,520,620 |
| 1914–1915 | 3,158,989 | 1,398,191 |
| 1913–1914 | 3,226,086 | 1,938,149 |
| 1912–1913 | 3,490,420 | 1,202,166 |
| 1911–1912 | 3,397,037 | 2,853,120 |
| 1910–1911 | 3,558,275 | 2,262,593 |
| 1909–1910 | 3,644,150 | 2,206,984 |
| 1908–1909 | 3,898,297 | 1,861,192 |
| 1907–1908 | 3,933,271 | 2,539,941 |
| 1906–1907 | 4,137,259 | 3,582,324 |
| 1905–1906 | 4,117,050 | 1,672,188 |
| 1904–1905 | 4,154,000 | 3,572,371 |
| 1903–1904 | 4,361,800 | 3,661,800 |
| 1902–1903 | 4,628,200 | 2,595,110 |
| 1901–1902 | 4,833,100 | 3,943,100 |
| 1900–1901 | 4,791,400 | 4,606,800 |

* Inclusive of 308,191 tons carried over from previous year.
† Inclusive of 499,170 tons carried over from previous year.

plants, 12.2 per cent had absorption, and 2.7 per cent had both systems. It may be mentioned that nearly every plant installed of late years has been either electric or internal combustion engine-driven, which means that few distilled water systems now go in, and it is doubtful whether a plate system plant has been installed in the last ten years in the United States.

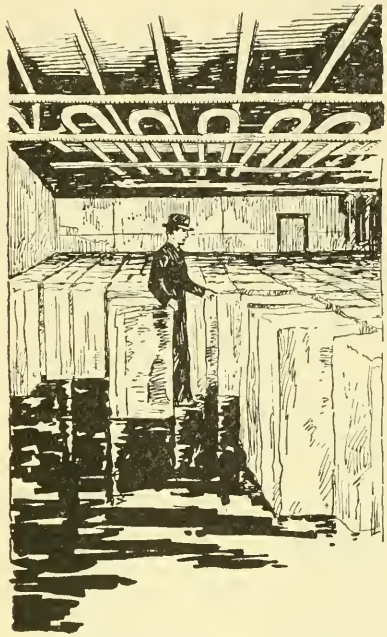


Fig. 248.—Daily Ice Storage.

The ice manufacturing industry, like all applications of refrigeration, has changed from steam engine to oil engine and particularly to electric motor-driven compressors. With the steam engine it was convenient to use the exhaust of the engine as the source of the water to be frozen, under which conditions the system became a distilled water can plant. With the rise in the cost of fuel and labor, and the desirability of the use of electric power—both from its simplicity and also from its cleanliness—there has been a very decided change, and few steam engines are now being installed in new plants except under special circumstances. This means, practically, the impossibility of using the distilled water system and therefore these electric or internal combustion engine-driven compressors are of necessity designed for the raw water can system. The use of electricity also has directly stimulated economies in the cost of ice production, and, in fact, is the greatest single factor in decreasing ice manufacturing costs.

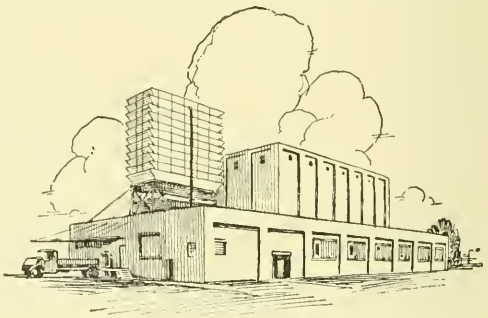


Fig. 249.—Typical Ice Making Plant.

**The Distilled Water System.**—If well, lake, river or pond water is frozen in cans without the proper system the ice produced will be marble-like in appearance, due to the air which is always in solution in the

water. This marble or opaque ice is good ice as far as any useful purpose is concerned. It melts at 32 deg. F., and its latent heat of fusion is practically the same as crystal ice, but the American public does not like it for table or household purposes. Where crushed ice is used, as in ice cream packing, ice and ice and salt railroad car icing, or in fish packing, there is no difference between the two, but for the retail trade the transparent ice must be supplied.

One of the first successful devices for the manufacture of crystal artificial ice is the distilled water system. The exhaust from the steam engine is condensed (at or near 212 deg. F.) and is brought up to a boil again while exposed to the atmosphere in order to permit the air present in the steam to pass out (Fig. 250.) This so-called reboiler has a skimmer or surface blowoff to remove the surface foreign material, as cylinder oil,

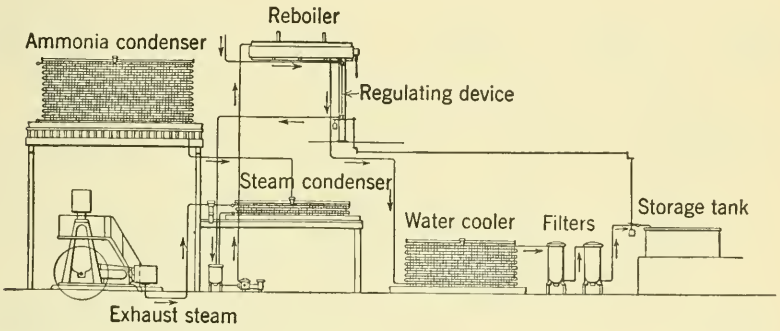



FIG. 250.—The Distilled Water Ice Plant.

etc. The condensate is then cooled by means of water-cooled coils to about 70 to 80 deg. F., and is passed through both a sand and a charcoal filter—the latter to remove any traces of oil present. The distilled water is now ready for the cans unless some system of precooling is available. The resulting ice, if care is taken not to let air become absorbed by the distilled water, will be transparent without the medium of any complicated auxiliary devices. Being free of air and dissolved salts, the water will freeze perfectly clear, unless impurities get into the water and cause a white or a red core. Figure 250 shows the ordinary layout for a distilled water can ice-making system.

There are certain localities where the distilled water plant has to be used. The so-called raw water can ice plant must have good water for the cans. If this water is heavily impregnated with salts of a nature that cannot be precipitated, for example, sodium salts, it is probable that the distilled water plant will have to be used. For such a plant the multiple-effect evaporator may work out satisfactorily (Fig. 251), or it is possible

that the old steam engine-driven compressor arrangement will be the best suited. The problem boils down to the best plan for freeing of the water from the salts in solution, and each case should be solved on its particular merits. While the distilled water system is not being installed in new plants to any great extent at the present time, there is a large number of plants that still use it. However, every year the number of plants changing over to electrically operated systems is large. One disadvantage of the distilled water system is the rapid deterioration of the distilling apparatus aside from the expense of the distillation.

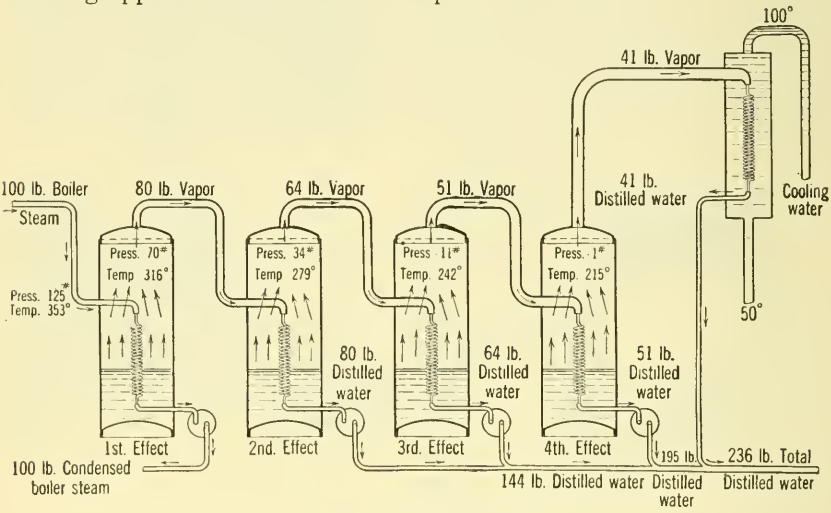



Fig. 251.—Four Effect Distilled Water Evaporator.

**The Raw Water System.**—One of the first of the so-called raw water systems (water not distilled is called raw water) was the *plate* ice method. This system is one in which the refrigerant maintains a plate at zero deg. F. or lower while it is submerged in a tank of water. Ice is formed on one side only, the water near the freezing surface being in a constant state of agitation. In the process of five or seven days a thickness of some 10 to 11 in. of ice is formed, and this ice is melted off the plate by permitting hot gas from the compressor to pass into the coils or the hollow part of the plate. The plate is quite large and the tank is deep, so that the cake of ice formed, which has to be lifted by a crane, weighs from three to five tons. This large cake is now removed to the saw table where 300-lb. cakes are made up.

The plate[2] system makes good ice, and, in fact, it is said that no

[2] Harry Sloan says (N.A.P.R.E., 1922) that the quality of plate ice has never been equalled by any other method of manufacture. Only slight differences in thick-

better ice can be made. It is non-uniform in thickness, however, and there is some prejudice against it on the part of the retailer. The tank is deeper than the can ice tank would be, and the head room has to be greater. The crane may have to be stronger, although the present tendency in can ice making is to increase the load on the crane, but the saw table involves expense for power and there is the trouble of snow ice formed by the sawing operation. The net result (due to the greater first cost, trade resistance and, as a rule, the greater operating cost) is an absence in the installation of new plate plants during the last few years.

**The Can Raw Water System.**—The history of raw water can ice making systems shows a large number of different devices. The attempt at first was to prevent opaque ice, due to dissolved air in the water, and later the problem became a desire to overcome the effects of dissolved salts as well as dissolved air in the water. Paddles were used in the early designs, and later a plunger with a pipe connection to the can so that an oscillation of the plunger created a surge of the water in the can. The entire tank, if a small one, has been placed on trunnions and has been rocked by a suitable mechanism. Experiments towards the elimination of all agitation by special treatment has been attempted for years and especially lately.[3] This latest process requires the use of galvanized piping throughout, a telescopic water storage tank, a special can filling device and finally a very careful chemical treatment and filtration followed by a subjection of the water heated to about 150 deg. F. to a very high vacuum. A second filtering completes the process. It is stated that the first and the operating cost is less than is required for high pressure air agitation.

Of all the methods of freezing water for commercial purposes the can system of raw water ice manufacture has proved itself to be the most successful arrangement. The cans are usually 300 or 400 lb., although smaller sizes are used in small tanks or where the ice is desired to be frozen during the day-time operation. The standard ice can size (Table 88) is about 11 in. by 22 in. cross-section and this size in the 300- or 400-lb. cake has been found to be convenient, both as regards freezing time and also as regards the handling and the cutting up for the trade.

In the can system the water to be frozen is put into the cans, and these are lowered into a tank of brine until the brine level (on the outside of the cans) is at least up to the surface of the water inside of the can.

ness and rounded edges appear on the top row of cakes. Much more evenness of thickness has been obtained since the introduction of the flooded system.

[3] Van R. Greene, Eastern Ice Association, April, 1923.

There is an advantage in having the cans submerged below this point, both as regards the shortening of the freezing time required and also as regards the decrease in the probable amount of checking of the ice. The brine for raw water ice is maintained cold at a temperature varying usually from 14 to 18 deg. F. by means of direct expansion piping in the tank or by means of shell and tube cooler submerged in the brine in the tank, and is kept circulated by means of a suitable propeller. The water to be frozen may have excessive salts, and always has a considerable amount of *air*, in solution. The former may be decreased, as regards the weight of actual solids, by the water softening treatment of lime and soda ash or by pumping out the core water after the can is about three-quarters frozen. These solids and the air in solution constitute a real problem, and their removal brings the variety of designs which are now to be found on the market.

TABLE 88

THE STANDARD SIZES OF ICE CANS

| Weight of Cake of Ice Pounds | Inside Dimensions, Inches | | | Length Overall, Inches | Thickness of Material, U. S. Standard Gage | |
|---|---|---|---|---|---|---|
| | Top | Bottom | Length | | Sides | Bottom |
| 25 | 4 × 9 | $3\frac{1}{2}$ × $8\frac{1}{2}$ | 23 | 24 | No. 18 | No. 18 |
| 50 | 5 ×12 | $4\frac{1}{2}$ ×$11\frac{1}{2}$ | 31 | 32 | No. 16 | No. 16 |
| 50 | 6 ×10 | $5\frac{1}{2}$ × $9\frac{1}{2}$ | 31 | 32 | No. 16 | No. 16 |
| 50 | 8 × 8 | $7\frac{1}{2}$ × $7\frac{1}{2}$ | 31 | 32 | No. 16 | No. 16 |
| 60 (25 Kilos.) | 5 ×14 | $4\frac{1}{2}$ ×$13\frac{1}{2}$ | 31 | 32 | No. 16 | No. 16 |
| 100 | 8 ×16 | $7\frac{1}{4}$ ×$15\frac{1}{4}$ | 31 | 32 | No. 16 | No. 16 |
| 200 | $11\frac{1}{2}$ ×$22\frac{1}{2}$ | $10\frac{1}{2}$ ×$21\frac{1}{2}$ | 31 | 32 | No. 16 | No. 16 |
| 200 | $14\frac{1}{2}$ ×$14\frac{1}{2}$ | $13\frac{1}{2}$ ×$13\frac{1}{2}$ | 35 | 36 | No. 16 | No. 16 |
| 300 | $11\frac{1}{2}$ ×$22\frac{1}{2}$ | $10\frac{1}{2}$ ×$21\frac{1}{2}$ | 44 | 45 | No. 16 | No. 16 |
| 400 | $11\frac{1}{2}$ ×$22\frac{1}{2}$ | $10\frac{1}{2}$ ×$21\frac{1}{2}$ | 57 | 58 | No. 14 | No. 14 |

**Air Agitation.**—The subject of air agitation has been one of careful investigation for some 10 or 15 years. It involves the chemistry of the water to be frozen, the efficiency of the filter process, the cost of power for agitation, the extra cost of the tank room work and the losses due to checking and chattering of the frozen cake. The appearance of the cake of ice is given much consideration so that the primary objective is a crystal cake with the least core or color.

The result has been designs of various sorts. The first system of agitation used involved a drop pipe into the center of each can about three-quarters of the way to the bottom of the can. Air at about 3 lb. pressure is applied and about 0.5 cu. ft. of free air per 300-lb. can is used. At a certain period in the freezing process the drop pipe is removed and the freezing is continued without agitation. Another method on the removal of the drop pipe is to suck the core and replace the water with distilled water, but this is not used in new plants at present, except in

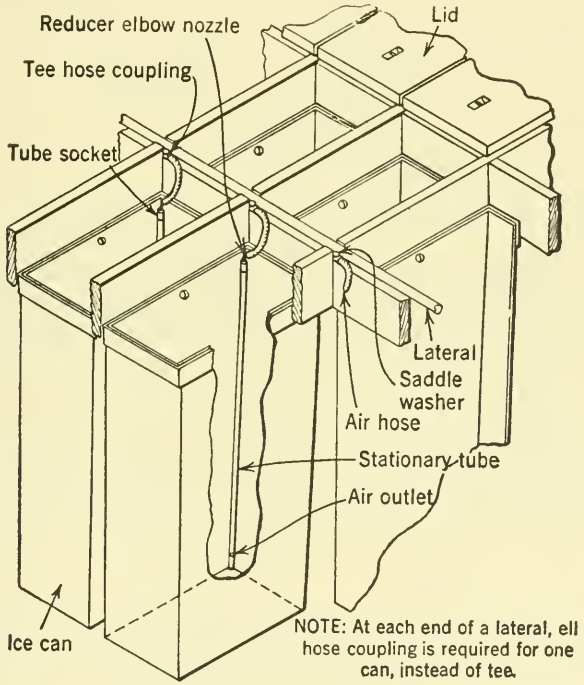



FIG. 252.—High Pressure Air Agitation.

very unusual cases. With this system, unless the drop pipe is removed at the proper time it will be frozen into the cake (requiring the thawing out by means of a steam needle), and its usefulness under these circumstances would be decreased because the pressure of 3 lb. is not great enough to prevent ice formation about the end of the pipe and the stopping of the agitation at that point. Finally, the can is lifted by a crane and is conveyed to the thawing platform where it is dipped into a tank of water at or about 70 deg. F. or in hot water (depending on local conditions) for an amount of time depending on special factors. When

the ice is loosened in the can it is placed on the tilting table and is permitted to slide on chutes into the daily ice storage room.

The low pressure air agitation[4] raw water system will give fairly good ice if good water is used. There is an expense due to the handling of the drop pipes, which must be removed at the proper time or be frozen into the ice, under which conditions the pipe has the additional expense of thawing by means of the needle, a process which results in added wear and tear to the drop pipe also. The low-pressure agitation drop pipe (Fig. 253) is placed in the center of the can and is *designed to be frozen*

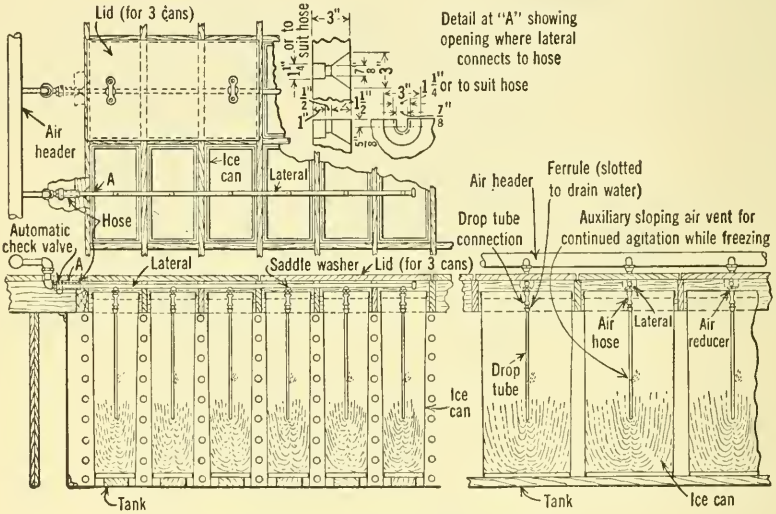



Fig. 253.—Low Pressure Air Agitation.

into the cake of ice. The seamless drawn brass tubing is perforated with small holes at several points to permit agitation of the core after the

[4] Some water causes considerable difficulty because the agitation is not vigorous enough to prevent impurities from settling to the bottom and forming a dirty streak in the ice. The cure would be either better softening (and filtration) or earlier dropping of the core, or agitation with a greater air pressure. But the low-pressure air system is much cheaper in first cost, and it is often used where the water is good. Harry Sloan says (N.A.P.R.E., 1922) that the disadvantages of low-pressure air agitation are thick opaque cores and greater labor costs. The advantages are in the low first cost, in more simple installation and most economical in the use of power.

H. R. Halterman (N.A.P.R.E., 1922) says that 100 cu. ft. of free air per minute compressed to 3 lb. per sq. in. pressure requires 2½ hp., and 0.5 cu. ft. per minute per can is used. He states also that 100 cu. ft. of free air per minute compressed to 30 lb. requires 10 hp., but that only 0.2 cu. ft. of air per minute is required per can.

end of the tube is filled up by the formation of ice. It may be mentioned that the present tendency seems to be to permit the pipe to freeze into the cake, and thereby permit more agitation to take place, decreasing by this means the amount of white core in the finished cake.

TABLE 89

Low-pressure Freezing System Apparatus—Air Blowers

| Tons of Ice | Number of Cans | Size of Pipe, Inches | C.f.m. | R.p.m. | Horse-power of Motor Required | Size of Blower Pulley, Inches | Belted Weight, Pounds | Maximum R.p.m. of Motor |
|---|---|---|---|---|---|---|---|---|
| 1 | 16 | 2½ | 8 | 520 | .75 | 7×1.5 | 90 | 1200 |
| 2 | 30 | 2½ | 15 | 635 | .75 | 7×1.5 | 90 | 1200 |
| 3 | 42 | 2½ | 21 | 600 | 1.5 | 8×2 | 200 | 1200 |
| 4 | 56 | 2½ | 28 | 670 | 1.5 | 8×2 | 200 | 1200 |
| 5 | 70 | 2½ | 35 | 740 | 2 | 8×2 | 200 | 1800 |
| 6 | 84 | 2½ | 42 | 810 | 2 | 8×2 | 200 | 1800 |
| 8 | 112 | 2½ | 56 | 700 | 3 | 10×2.5 | 300 | 1800 |
| 10 | 144 | 2½ | 72 | 800 | 3 | 10×2.5 | 300 | 1800 |
| 12 | 168 | 3 | 84 | 505 | 5 | 12×3 | 400 | 1800 |
| 15 | 210 | 3 | 105 | 550 | 5 | 12×3 | 400 | 1200 |
| 20 | 280 | 3 | 140 | 660 | 5 | 12×3 | 400 | 1200 |
| 25 | 360 | 3 | 180 | 790 | 5 | 12×3 | 400 | 1200 |
| 30 | 420 | 4 | 210 | 460 | 7.5 | 16×4 | 730 | 1200 |
| 35 | 484 | 4 | 242 | 505 | 7.5 | 16×4 | 730 | 1200 |
| 40 | 560 | 4 | 280 | 565 | 7.5 | 16×4 | 730 | 1200 |
| 50 | 720 | 4 | 360 | 680 | 7.5 | 16×4 | 730 | 1200 |
| 60 | 840 | 4 | 420 | 460 | 7.5 | 16×4 | 1460 | 1200 |
| 80 | 1120 | 4 | 560 | 565 | 7.5 | 16×4 | 1460 | 1200 |
| 100 | 1440 | 4 | 720 | 680 | 7.5 | 16×4 | 1460 | 1200 |
| 120 | 1680 | 6 | 840 | 500 | 10 | 20×4 | 3000 | 1200 |
| 150 | 2160 | 6 | 1080 | 550 | 15 | 20×4 | 3000 | 1200 |
| 200 | 2880 | 6 | 1440 | 550 | 15 | 20×4 | 4500 | 1200 |

Blowers operate at 3 lb. gage pressure.

Very seldom is the water to be frozen free from some trace of solids. The saying that water which is good enough to drink is good enough for raw water ice needs some reservations (Fig. 254.) The solids concentrate

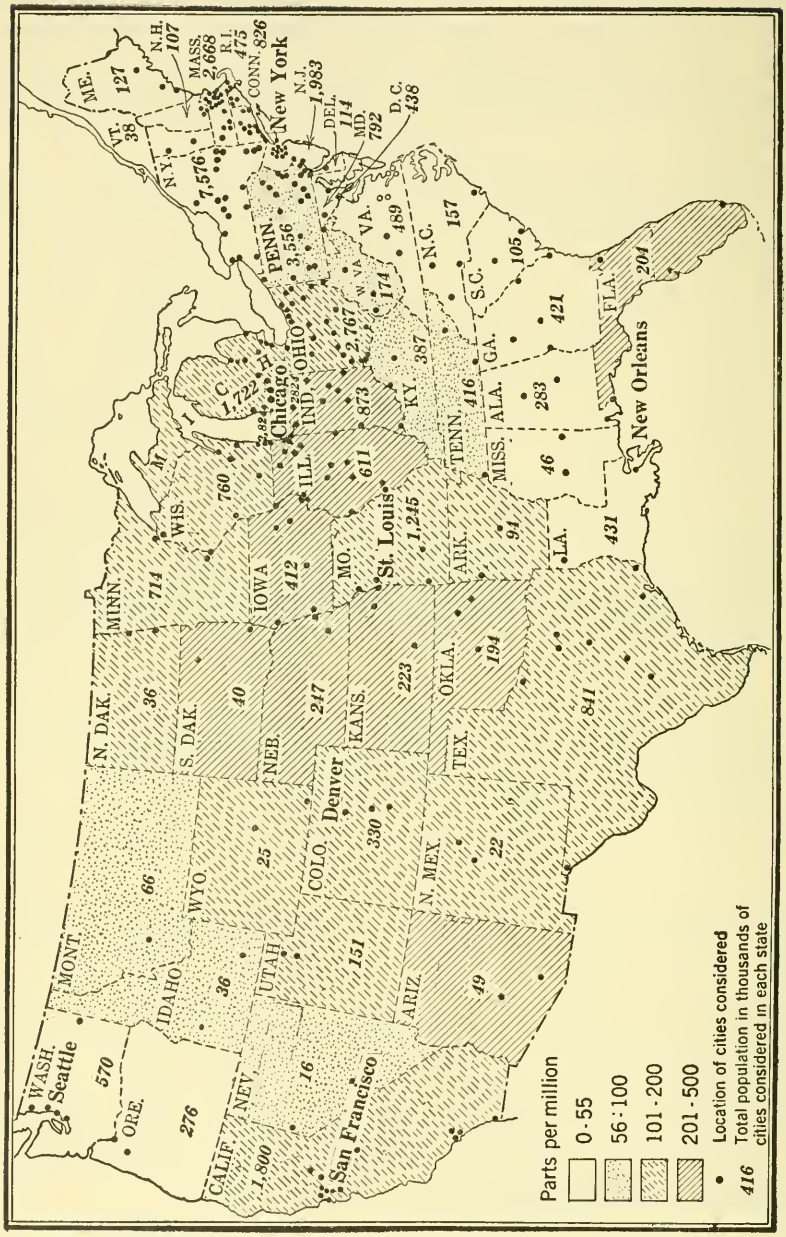



FIG. 254.—Hardness of Well Water in the United States.

as freezing continues, and the core water becomes very briny to the taste. The core needs to be removed once and sometimes twice in the 300- and 400-lb. cake, both from the viewpoint of the foreign material and also because of the reduced freezing temperature of the core water and the increased time of freezing required. This is done by means of a core sucker (Fig. 265). The air for agitation is best taken from the top of the tank, thereby getting a colder air than would be available in the summer from the atmosphere and a lower moisture content. The air is delivered, as a rule, by means of large headers to the laterals. The air pressure is therefore more uniform all over the tank.

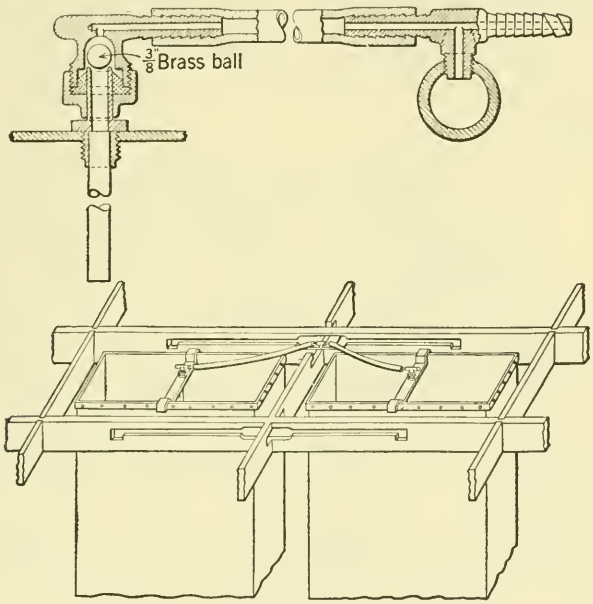



Fig. 255.—Medium Pressure Can Agitation—Center Tubes

**The High and the Medium Air Pressure.**—If the drop pipe is located in the center of the can it is subjected to 32 deg. F. until the very last part of the freezing of the can, and under these conditions no trouble is experienced from the freezing of the moisture content of the air. However, if the drop pipe is made an integral part of the can, either soldered in the corner (Fig. 252) or fastened to the side of the can, it comes into metal contact with the brine, which is at 14 to 18 deg. F. Such air agitation, spoken of as high pressure or medium pressure air, will need to be dehumidified in order to reduce the dew point of this air. As air under 10 to 20 lb. is required at the laterals, the compressor usually

supplies air at a pressure greater than this by an amount varying from a few pounds up to 30 lb. more than is required at the cans.[5] Some manufacturers have no high pressure, but what is called a medium pressure air system (Fig. 255). In another form the drop pipe is soldered into the corner of the can, and a hole in the side of the pipe about 6 in. from the bottom of the can permits the air to come out. However, dehumidifiers are required in all cases except the low-pressure systems, and there are three forms of these in general use.

**The Dehumidifier.**—The first form employs a cylindrical shell one-half full of water and kept cool with brine from the brine tank. The air is admitted near the bottom by passing through pipes or plates perforated with small holes, and as it passes upwards through the water, is washed, cooled and purified. Apparently much depends on the design of the air openings as it is possible for the air to pass in streams instead

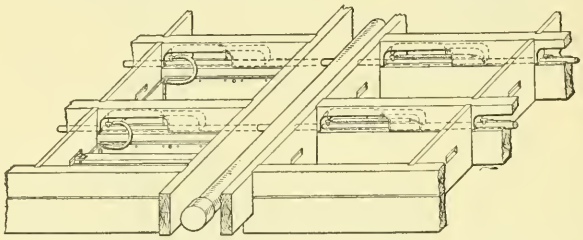


Fig. 256.—Medium Pressure Can Agitation—Corner Tubes.

of as bubbles under which conditions the efficiency of the dehumidifier will decrease. Finally, the air is further cooled and dehydrated by means of a brine coil.

The second type uses two cylindrical shells through which the air passes in series, the first being one-half full of water cooled by means of a brine coil and the second half-full of brine which is cooled by the use of direct-expansion piping. The air line to the cans has also usually

[5] For 18 lb. air a hole made by a No. 75 drill is satisfactory or a No. 53 drill and a No. 1f B. and S. gage wire. The air should be compressed to 25 to 30 lb. at the compressor if 14 to 16 deg. F. brine is to be used.

The high-pressure air system has the advantage because of the necessity of the removal of the air tube in the low-pressure system, with the resulting stoppage of agitation and the large core in the ice block as well as the labor required in the work. The high-pressure air system requires the use of a constricting orifice.

Thomas Shipley says that the low-pressure air system is all right with proper care and some waters, but the high-pressure air system would produce much better results on the average than will the low-pressure even when the low-pressure agitation is doing its best as there is no stoppage of agitation at any time; and that the core spaces do not require washing out except when the sediment is very heavy.

a cartridge filter which has to be recharged every four hours. A modification of the method uses a brine spray (Fig. 257).

The third type is composed of two shell and coil or shell and tube cylindrical tanks, brine cooled,[6] through which the air passes in series. The second tank collects a frost on the cooling surfaces and must be defrosted every six to eight hours.[7] This is done by the simple device of reversing a four-way valve. The particular advantage of this type is its simplicity, the lack of care required in its operation, the avoidance of dilution of the brine as is the case where the air comes in direct contact with the brine and the lack of any need of filtering in order to collect the salt held in the air which would tend to close the needle valves in the can pipes. The frost accumulation on the pipes assists in cooling the air by an amount at least equal to the latent heat of fusion of the ice when the four-way valve is reversed (Fig. 258).

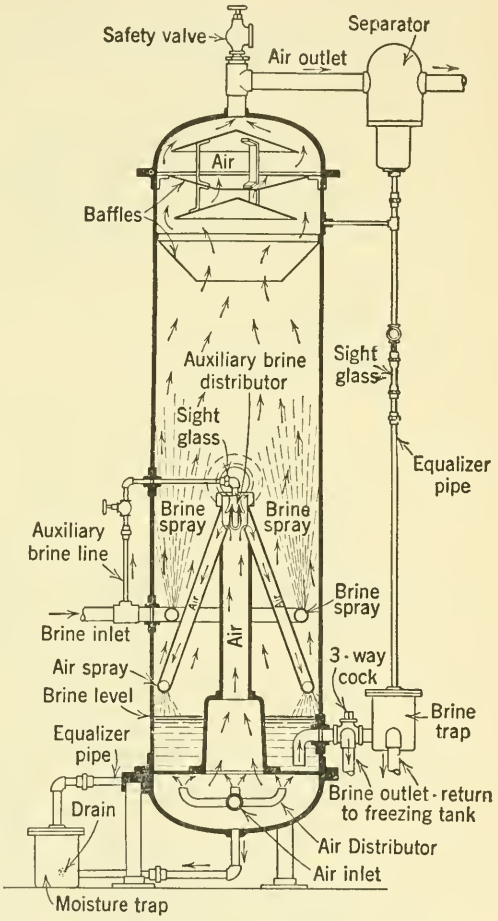



FIG. 257.—Air Dehydrator, Using Brine Sprays.

The use of air agitation involves considerable operating expense

[6] Shipley says that the use of brine instead of direct expansion for air cooling is preferred because if the compressor stops at any time the air can be kept cold and therefore dehumidified. Air pressures of 5 lb. for low pressure and 25 lb. for high pressure use the same motor for either condition and therefore the same power.

[7] The possible objection to this is the accumulation of frost on the second brine coil which soon decreases the efficiency of the cooling surface to a point where cooling below 32 deg. F. would be difficult, unless the four-way valve is operated as well as the brine valves.

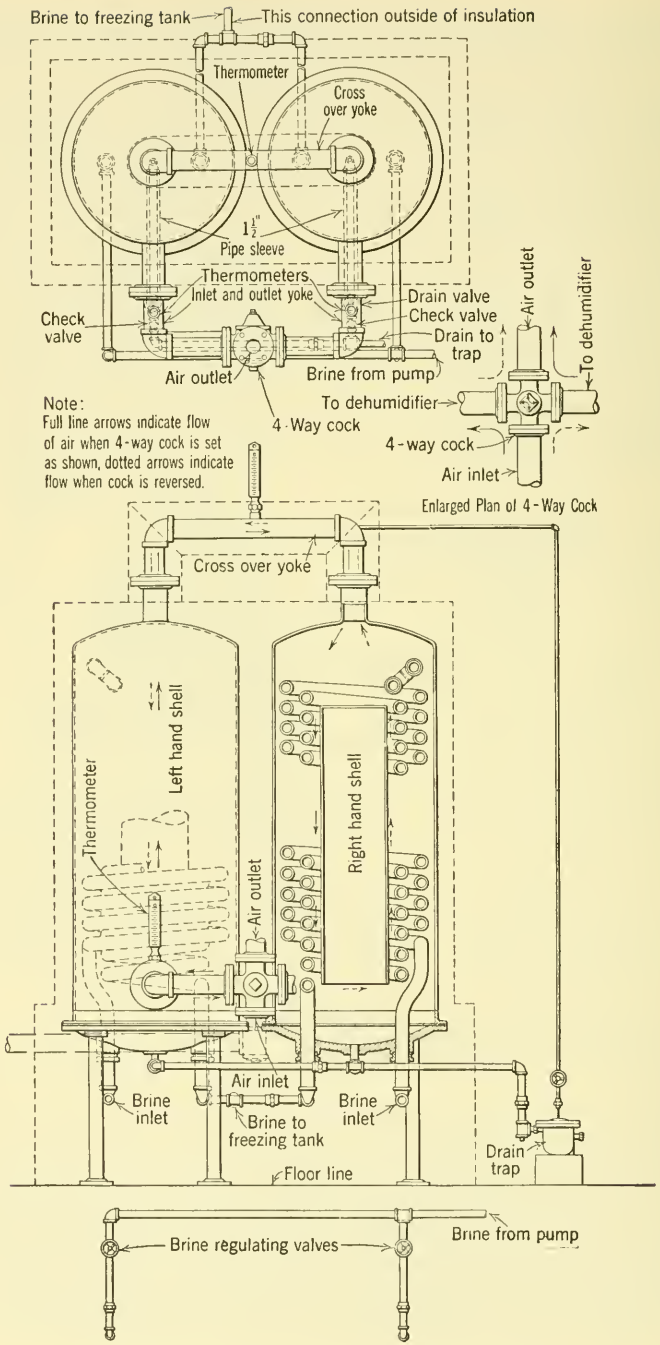



Fig. 258.—Air Dehydrator—Using Brine Coils.

and investment. The medium air pressure system requires slightly less air than the low-pressure agitation and a little more power, but the ice quality is better. It is claimed that the high agitation[8] system requires 6 kw. hr. per ton, or approximately one-eighth of the entire power required of the plant. One man is used in the tank room per shift up to 60 tons output per 24 hours. The ratio of the power requirements is about 1.0 hp. per 100 300-lb. cans for low-pressure air, and from 4.0 to 5.0 hp. per 100 300-lb. cans for high-pressure air agitation, and in addition there is required the refrigeration necessary to cool the air, and the power for the water and the brine pumps. About $\frac{1}{2}$ hp. is used per 100 300-lb. cans for the core pump. If air agitation could be eliminated, it would be a great means of simplifying the manufacture of ice, and attempts are being made in that direction. The modern tendency (1927) is towards the use of a medium-pressure air (about 10 lb.) using a drop pipe in the center of the can, made of brass and extending nearly to the bottom of the can (usually about 5 in. of the bottom) as compared with about 9 in. for the low-pressure system. It gives a cake of ice with hardly any perceptible core, saves tank room labor and does not require much more power than the rather crude low-pressure drop pipe system of the earlier designs.

**Ice Dumping and Refilling the Cans.**—The tank room routine consists in the filling of the cans with water, the freezing and, finally, the dumping of the ice. The usual method is to lift a number of cans by means of the can hoist and to convey them to the dump where the can filler is usually located. Much of this work is heavy routine, and it takes almost as long to lift two cans as it does to lift six or eight. To reduce the cost of labor in the tank room 2-can hoists are now limited as a rule to very small ice tanks, but the larger tanks have cranes which will lift as many as 24 to 30 cans at a time, using the heavy 3-motor crane. Heretofore the connection from the cans to the crane has been by means of can dogs, but when four, six or more cans were "lifted" considerable time was wasted because of the undue swinging of the cans at the start and the stop, and injury to the cans or the direct expansion piping in the tank. The result has been the use of the so-called can basket—a device which means simply the assembling together of all the cans to be lifted at one time, and a means of attaching the crane chains to the basket. Suitably held together a row of 24 cans can be manipulated nearly as quickly as the old 2-can hoist. The basket is

[8] Sloan says (N.A.P.R.E., 1922): Every hp. additional added to auxiliaries of the plant mean $\frac{1}{4}$ kw.-hr. per ton ice (allowing for efficiency of motors, etc.). Whyte says in reply that cost of power in five plants (Consumers Ice Co., Chicago) checked exactly with amount used in low-pressure plants.

sometimes made of an angle iron frame, and the additional weight assists in submerging the cans in the brine and also decreases the necessary strength of the tank framing, and permits the cans to be placed on closer centers. In some designs the use of baskets requires, however, the placing of the piping on the *bottom of the tank* or its elimination entirely by the use of the shell and tube brine cooler. The saving of time in the tank room has been found to be about 30 per cent using baskets as compared with can dogs. With the decreased size of the tank frame by using the basket it has been found that less tank room, roof and crane space are required, less ground space, insulation and foundation work, less framework and brine charge, fewer accidents and less

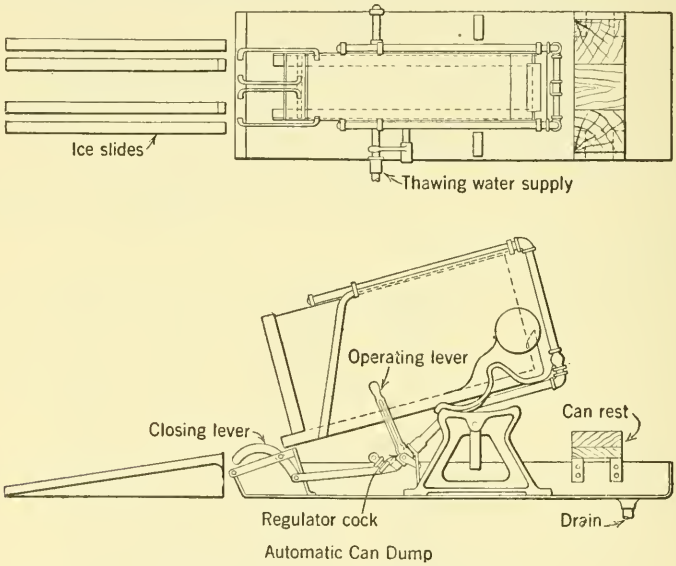



Fig. 259.—Automatic Can Dump.

damage to the cans and piping, etc., resulting in 10 per cent less space and 50 per cent less repairs (Ralph Cole, N.A.P.R.E., 1923).

Tests of ice tanks (R. C. Doremus, N.A.P.R.E., November, 1922) showed considerable variation in the operating conditions when using large baskets. With 960 400-lb. cans, using $38\frac{1}{2}$ diam. by 14 ft. long shell and tube brine cooler and 2 18-in. propellers at each end, pulls of 12 cans caused a brine temperature rise of 0.7 deg. F., whereas 16 cans at a time caused a temperature rise of 1.0; 20 cans, 1.6, and 24 cans caused a temperature rise of 1.9 deg. F. Lifting 8 cans at a time caused the brine level to fall $1\frac{1}{4}$ in., and 24 cans caused the brine level to drop $5\frac{1}{4}$ in. These changes of brine level would require special designs. It is

claimed that a can lift of 1 to $1\frac{1}{4}$ per cent is the maximum amount permissible at a time unless an excessive variation of the operating conditions is to be permitted.

**The Dip Tank.**—The old-fashioned sprinkler dump (Fig. 259) is now being replaced by the dip tank. One of the operating difficulties in ice making is that of the checking of the ice. Checking is due to so

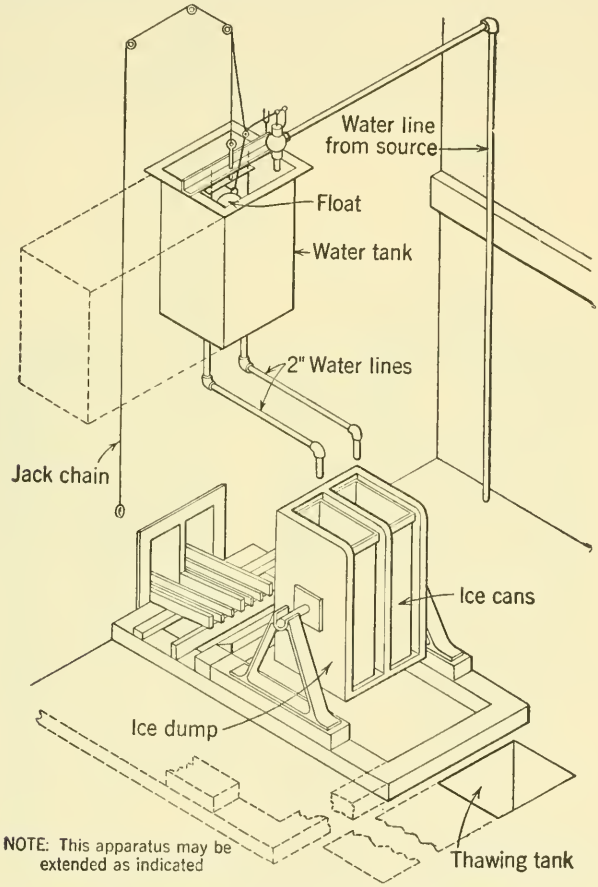



Fig. 260.—Automatic Can Filler.

many factors that operation in any two plants is seldom alike. Some plants use 60 deg. F. water in the dip tank and others have the water steaming. The present tendency is to have the dip tank water agitated with air (in the case of medium and high-pressure systems) in order to give greater heat transfer to the cans and to stimulate a more even temperature all around the cake of ice, through destroying stratification

by removing the cold film of water about the cans. A gain of 42 per cent in the time required for thawing of the ice is claimed for the air-agitated dip tank.

**Can Filling.**—Even in the small ice-making plants it is doubtful whether the old float valve type of automatic can filler is profitable. It is slow, expensive to operate, and unless carefully operated may result in a weakened brine. At present the dip tank, the can dump (which is simply a rest for the cans on trunnions) and the measuring tank type of can filler are designed as a unit (Fig. 260). With the modern type of can filler the correct amount of water is ready in each compartment (there is

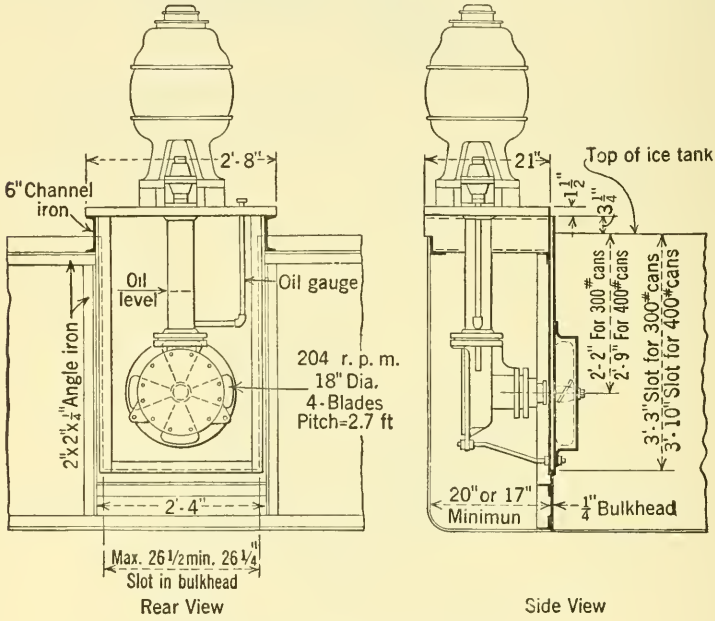



FIG. 261.—Vertical Brine Agitator.

a separate compartment for each can) and a pull of the control chain starts the filling operation. With a 3-in. filler pipe the 400-lb. can can be filled in 26 seconds, whether 2 or 20 cans are filled at the same time. While the tank man is pulling another batch of cans the filler tank is receiving another charge of water.

The cans, filled with water, will submerge quickly to the proper level without difficulty and with less wear and tear to the equipment. This is especially important of late years because of the amount of agitation given the brine. This would tend to carry the can out of plumb unless they were restrained by the means of can guides. The dump, for the

larger plants, is electrically operated by means of cables driven by a slow speed electric dump hoist.

**Agitation.**—The brine must be agitated (Figs. 261 and 262 and Table 90) in order to secure results of any value in the ice tank. This is because of the tendency for stratification and the formation of films about the ammonia piping and the ice cans. The advent of raw water ice has

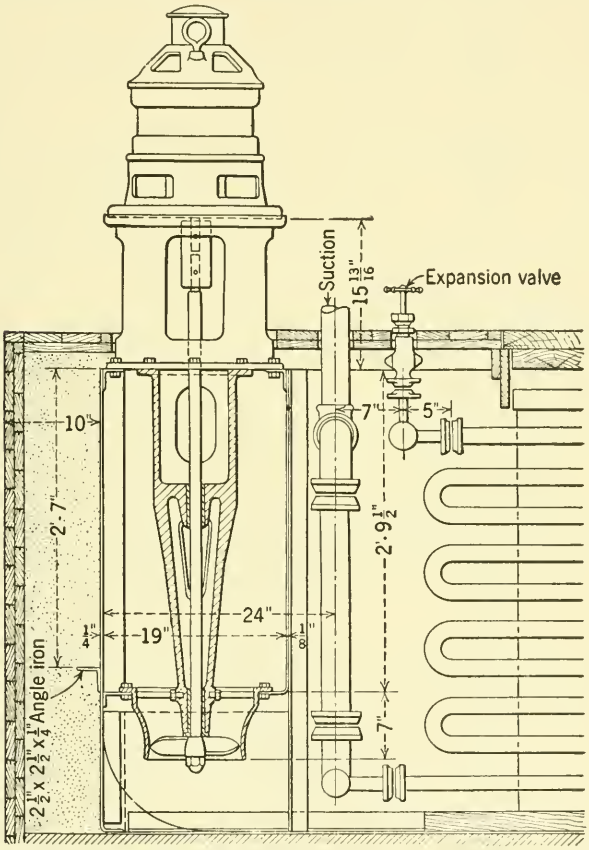



Fig. 262.—Vertical Brine Agitator.

changed conditions considerably in this respect. Raw water ice can be made, as a rule, only with 14 deg. F. brine or higher, because of the tendency for checking. It is also uneconomical to operate an electrically driven compressor with low-temperature brine. The net result is a striving for a close temperature range between the temperature of the ammonia boiling in the pipes and of the brine.

TABLE 90

STANDARD FREEZING TANK PROPELLERS

| Capacity, Tons | Horizontal | | | | | | | | | | Vertical | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | Motor | | Motor pulley | | | | | | Motor | |
| | Number | Diameter, inches | Pitch, inches | Revolutions | Rubber belt, inches | Pulley, inches | Horsepower | R.p.m. | Diameter, inches | Face, inches | Number | Diameter, inches | Pitch, inches | Revolutions | Horsepower | R.p.m. |
| 1 | 1 | 12 | 6.0 | 103 | 3 | 34 | | | | | 1 | 7 | 6 | 850 | | |
| 2 | 1 | 12 | 6.0 | 125 | 3 | 28 | 1 | 1160 | 3 | $3\frac{1}{2}$ | 1 | 7 | 6 | 850 | 2 | 850 |
| 3 | 1 | 12 | 6.0 | 145 | 3 | 24 | | | | | 1 | 7 | 6 | 850 | | |
| 4 | 1 | 15 | 12.0 | 145 | $3\frac{1}{2}$ | 32 | | | | | 1 | 9 | 6 | 850 | | |
| 5 | 1 | 15 | 12.0 | 165 | $3\frac{1}{2}$ | 28 | 2 | 1160 | 4 | $4\frac{1}{2}$ | 1 | 9 | 6 | 850 | 3 | 850 |
| 6 | 1 | 15 | 12.0 | 180 | $3\frac{1}{2}$ | 26 | | | | | 1 | 9 | 6 | 850 | | |
| 8 | 1 | 15 | 22.0 | 145 | 4 | 32 | | | | | 1 | 9 | 9 | 850 | | |
| 10 | 1 | 15 | 22.0 | 165 | 4 | 28 | 3 | 1160 | 4 | $4\frac{1}{2}$ | 1 | 9 | 9 | 850 | 3 | 850 |
| 12 | 1 | 15 | 22.0 | 180 | 4 | 26 | | | | | 1 | 9 | 9 | 850 | | |
| 15 | 1 | 15 | 22.0 | 193 | 4 | 30 | | | | | 1 | 12 | 12 | 850 | | |
| 20 | 1 | 22 | 39.9 | 152 | $4\frac{1}{2}$ | 38 | 5 | 1160 | 5 | $5\frac{1}{2}$ | 1 | 12 | 12 | 850 | 5 | 850 |
| 25 | 1 | 22 | 39.9 | 160 | $4\frac{1}{2}$ | 36 | | | | | 1 | 12 | 12 | 850 | | |
| 30 | 2 | 22 | 39.9 | 130 | 5 | 36 | | | 4 | | 2 | 9 | 9 | 850 | | |
| 35 | 2 | 22 | 39.9 | 153 | 5 | 38 | $7\frac{1}{2}$ | 1160 | | $5\frac{1}{2}$ | 2 | 9 | 9 | 850 | 3 | 850 |
| 40 | 2 | 22 | 39.9 | 153 | 5 | 38 | | | 5 | | 2 | 12 | 12 | 850 | | |
| 50 | 2 | 22 | 39.9 | 180 | 6 | 38 | 10 | 1160 | 6 | $6\frac{1}{2}$ | 2 | 12 | 12 | 850 | 5 | 850 |
| 60 | 2 | 22 | 39.9 | 180 | 6 | 38 | | | | | 2 | 12 | 12 | 850 | | |

In order to get this close operation and at the same time to acquire production a heavy agitation is necessary. Some tanks have been designed for a difference in the brine level of from three to five or more inches in the tank. There is also more thought in regard to the regulation of the velocity of the brine in different parts of the tank. With a tapering can it is evident that there is more space at the bottom of the brine tank between the cans for the brine to pass through than there is at the top. Also there is usually considerable space at the bottom of the tank below the cans. Since for best results the brine flow should be reasonably uniform around the cans, attention is now being directed to this matter.

The amount of agitation is not at all uniform, as has been mentioned, but the following is the practice among some engineers: for 400 cans or less use one agitator, and for 24 cans wide and about 20 cans up to 35 cans long use two agitators, at the same end of the tank. For more than 24 cans wide and more than 35 cans long use *two* agitators at *each* end. The power required of the agitators is approximately $2\frac{1}{4}$ hp. per agitator, propeller 204 r.p.m., 18 in. diam., 4 blades, 2.7 ft. pitch.

**Water Softeners.**—*Deep well water* varies in the amount of salt in solution in the different parts of the United States (Fig. 254). The eastern seaboard has good water except in Florida, where it is very bad. The central states have hard water, as for example, Illinois, Indiana, Iowa, South Dakota, Nebraska, Kansas and Oklahoma. Washington and Oregon have soft water, but California water is bad. The water of the Great Lakes is hard. In the southwest the water in certain regions contains so much sodium bicarbonate as to make it unfit for raw water ice making.

If water contains more than a trace of solids in solution before it is suitable for can ice manufacture it must have a special treatment if a crystal cake of ice is to be secured, for as freezing continues the mineral impurities are separated out as a precipitate and will show as a white or as a colored deposit. Thus magnesium salts will cause a white and iron salts a red deposit in the ice. If agitation is not resorted to these particles as well as the air in solution in the water will settle on the surface of the ice and being frozen into the ice will give the peculiar opaque appearance which is not desired by the retail trade. If the water is agitated, the impurities will be kept moving in the core water, and this action will continue until the core water gets more and more briny until settlement will occur or the core is removed, to be replaced with fresh water (or under exceptional circumstances with distilled water). If the core was a large one requiring a large replacement, it is possible that a second core pulling would be required. In addition to

Calcium carbonate is limestone ($CaCO_3$). Water ($H_2O$) as rain, absorbs carbon dioxide ($CO_2$) as it falls to earth, and from the earth, and then the water with the absorbed $CO_2$ dissolves the limestone. The chemical equation is written as follows:

$$CaCO_3 + CO_2 + H_2O = CaCO_3 \cdot CO_2 \cdot H_2O$$

Limestone + Carbon + Water = Calcium Bi-carbonate
(Insoluble) Dioxide (Soluble)

The same is true of magnesium carbonate. But water chemists have reported the bi-carbonates of calcium and magnesium in terms of calcium carbonate and magnesium carbonate for so long that it has become standard practice. Occasionally, however, an analysis is submitted where they are expressed as bi-carbonates, and the following factors may be of help:

Calcium Bi-carbonate × 0.617 = Calcium Carbonate.
Calcium Carbonate × 1.62 = Calcium Bi-carbonate.
Magnesium Bi-carbonate × 0.576 = Magnesium Carbonate.
Magnesium Carbonate × 1.735 = Magnesium Bi-carbonate.

The following table gives the names of the usual minerals found in water, how they effect the appearance of raw water ice, and the effect of treatment with hydrated lime in an efficient water treating plant.

| Minerals in Water | Effect in Ice | Result of Treatment With Hydrated Lime $Ca(OH)_2$ |
|---|---|---|
| Calcium carbonate | Forms gritty, dirty, discolored deposit, usually in lower part and center of the cake. Causes shattering at low freezing temperatures. | Practically eliminated. |
| Magnesium carbonate..... | Forms gritty, dirty, discolored deposit, milky patches and bubbles, and also causes shattering at low freezing temperatures. | Practically eliminated. |
| Iron oxide..... | Causes bad discoloration, yellow or brown deposits and also stains the calcium and magnesium deposits. | Eliminated. |
| Aluminum oxide and silica..... | Cause dirty deposit and sediment. | Practically eliminated. |
| Suspended matter | Causes dirty deposit and sediment. | Eliminated. |
| Calcium sulphate calcium chloride | Act like and are no worse than sodium sulphate and sodium chloride. Do not form deposit. | No change. |
| Magnesium sulphate..... | Cause greenish or grayish cast, concentrate in core water, retard freezing and cause heavy cores. Often show up as white spots and dirty colored streaks. Also act like sodium sulphate and sodium chloride. Do not form deposit. | Changes to calcium sulphate. |
| Magnesium chloride..... | | Changes to calcium chloride. |
| Sodium sulphate, sodium chloride. | Cause white butts, concentrate in core, make heavy core and retard freezing—but do not form deposit. | No change. |
| Sodium carbonate (actually present as sodium bi-carbonate).. | In only small quantities often causes shattering at temperatures below 16 degrees. Also causes white butts, concentrates in core, retards freezing and makes heavy cores. Does not form deposit. | Treatment changes sodium bi-carbonate to sodium carbonate —treatment improves but little. |

| Minerals | Allowable Limits (With Remarks) |
|---|---|
| Calcium carbonate and magnesium carbonate (combined)..... | If 70 p.p.m. (4 grs. per gallon)—or less—lime treatment required but *alum* treatment recommended to change the carbonates to less objectionable calcium sulphate and magnesium sulphate.<br>If total exceeds 70 p.p.m. (4 grs. per gallon)—lime treatment must be used to prevent objectionable deposit. Calcium and magnesium carbonates can be present in any quantity—lime treatment will remove all but 30 or 40 p.p.m. (1.8 or 2.4 grs. per gallon). |
| Iron oxide..... | If more than 0.2 p.p.m. (0.012 gr. per gallon) treatment is required. This trace of iron may be sufficient to badly discolor deposits of calcium and magnesium that otherwise might be unobjectionable.<br>Some analyses report the iron and aluminum oxides together. Such analyses are valueless because iron is so objectionable that its quantity and form should be fully understood. |
| Aluminum oxide and silica..... | Not usually present in quantities which alone would require treatment. |
| Suspended matter... | If only objectionable impurity in the water, filtration is the only treatment required. |
| Color..... | Analyses may indicate the presence of color, but not show to what color is due. Surface waters, low in dissolved mineral matter, but with color, should be given expert individual consideration, as color may or may not be easily removed. |
| Calcium sulphate....<br>Magnesium sulphate.<br>Calcium chloride....<br>Magnesium chloride..<br>Sodium sulphate.....<br>Sodium chloride.....<br>Sodium carbonate... | All the sulphates, chlorides and the sodium carbonate have, in general, the same effect on the appearance of raw water ice and, therefore, in this rough tabulation, can be grouped together.<br>Generally, the sodium carbonate can be considered $1\frac{1}{4}$ times as objectionable as the sulphates, and the chlorides about $\frac{3}{4}$ as objectionable as the sulphates. Therefore total the sulphates and chlorides separately and apply the factors:<br>Total sulphates $\times 1.0$ = .....<br>Total chlorides $\times 0.75$ = .....<br>Sodium carbonate $\times 1.25$ = .....<br>Sum Total = .....<br>If sum total is less than 171 p.p.m. (10 grs. per gallon) first quality ice can be expected with efficient air agitation and average brine temperatures.<br>If sum total is more than 171 p.p.m. (10 grs. per gallon) and less than 256 p.p.m. (15 grs. per gallon)—good ice can be expected with slightly higher than average brine temperatures or slightly more than the usual volume of air for agitation.<br>If sum total is more than 256 p.p.m. (15 grs. per gallon) and less than 342 p.p.m. (20 grs. per gallon)—merchantable ice can be made if much more than the usual volume of air per can per minute is provided, especially during the beginning of the freeze and brine temperatures of 18 degrees or more are carried.<br>If sum total is more than 342 p.p.m. (20 grs. per gallon) and less than 684 p.p.m. (40 grs. per gallon)—the analysis or another sample of water should be submitted, together with full details of operating conditions, for a complete report as to whether an acceptable quality of ice can be made.<br>If sum total is more than 684 p.p.m. (40 grs. per gallon)—merchantable ice cannot be expected. |

| Name | Symbol | Reacts With | Becomes |
|---|---|---|---|
| Calcium bicarbonate..... | $(CaH_2)(CO_3)_2$ | $Ca(OH)_2$ | $\underline{2CaCO_3} + 2H_2O$ |
| Calcium sulphate..... | $CaSO_4$ | $Na_2CO_3$ | $\underline{CaCO_3} + Na_2SO_4$ |
| Calcium chloride..... | $CaCl_2$ | $Na_2CO_3$ | $\underline{CaCO_3} + 2NaCl$ |
| Magnesium bicarbonate... | $MgH_2(CO_3)_2$ | $Ca(OH)_2$ | $\underline{MgCO_3} + Na_2SO_4$ |
| Magnesium sulplate..... | $MgSO_4$ | $Ca(OH)_2$ and $Na_2CO_3$ | $MgSO + Na_2CO_3 = \underline{MgCO_3} + Na_2SO_4$<br>$MgCO_3 + Ca(OH)_2 = \underline{Mg(OH)_2} + \underline{CaCO_3}$ |
| Magnesium chloride..... | $MgCl_2$ | | $MgCl_2 + Na_2CO_3 = \underline{MgCO_3} + 2NaCl$<br>$MgCO_3 + Ca(OH)_2 = \underline{Mg(OH)_2} + \underline{CaCO_3}$ |
| Iron bicarbonate..... | $FeH_2(CO_3)_2$ | $Ca(OH)_2$ | $FeH_2(CO_3)_2 + Ca(OH)_2 = \underline{FeO} + CO_2 + \underline{CaCO_3} + 2H_2O$<br>$2\underline{FeO} + O_2 = \underline{Fe_2O_3}$ |

The underlined substance is insoluble.

the poor appearance of these salts in the ice, their presence has the effect of increasing most decidedly the amount of checking of the ice, resulting in a direct loss unless there is a market for crushed ice.

The treatment for the softening of water is quite simple. Certain salts cannot be removed, as, for example, the sodium compounds, whereas the calcium and magnesium salts usually can be precipitated. The usual method is to use lime ($Ca[OH]_2$) or lime and soda ash ($Na_2CO_3$). If the sum of the sulphates, chlorides and nitrates plus the sodium carbonate does not exceed 20 grains per gal. the water can be purified for ice making. From 20 to 40 grains it is doubtful, and over 40 grains per gal. gives water that is not satisfactory for ice making at all and may not be used for raw water ice making.

Up to 12 grains per gallon of sulphates, chlorides, nitrates and sodium carbonates (altogether) the low-pressure air system will answer. If high-pressure air is used at the bottom of the can this sum may be increased to 20 grains and the ice will look as well as 12-grain per gallon water with low-pressure agitation.

With 0.02 grain "iron oxide" per gallon removal is necessary or color will appear in the ice, especially as the ice appears to magnify any foreign material in the ice itself. It has been found that sodium salts particularly require agitation at the bottom of the can.

Treating the water always decreases the amount of core pumping necessary, and reduces in proportion the care required of the attendant on the can floor. The core becomes more and more concentrated as freezing continues. Calcium and magnesium carbonate freeze into the ice earlier than any other salts, being precipitated because of the loss of the carbon dioxide gas which held them in solution as bicarbonates, and this is hard to pump out unless a large core is pumped. These carbonates if frozen into the ice will remain in the ice box when the ice is melted.

Generally speaking, mineral matter begins to freeze into the ice when the concentration becomes from 55 to 60 grains per gallon. Ice made from hard water is not so firm, hard and crystal-like as if the water were treated, but is likely to be brittle and is far more likely to shatter. The presence of bicarbonates results in small white specks or bubbles frozen into the ice.

Certain water requires distillation, and water heavy in sodium sulphate, sodium carbonate, etc., is likely to have a white shell form around the bottom of the cake.

The treatment with lime and soda ash necessarily *takes place cold*, under which conditions the reactions take place slowly and incompletely. The usual method is to permit the water and the chemical to stand for four hours in the water softener settling tank. During the sedimentation

period the chemicals sink to the bottom where they are drained off as a thin sludge, and the softened water is drawn off near the top and is passed through a quartz filter to remove the free iron, silica and the suspended dirt and sludge.

The tendency of the design of water softeners is to give more time to sedimentation, about 6 to 8 hours in fact, thus securing more complete chemical reaction and settling. The filter is being made larger and is figured on the basis of a maximum of 3 gal. per sq. ft. per minute of cross-sectional area, and 12 gal. per min. per sq. ft. for backwash. There is also an alum coagulator to improve the filter action, although some plants add the alum in the downtake of the softener.

The muddy appearance of the ice in the spring and autumn may be caused by dirty water or it may be by the intermittent action of the filters during periods of light loads. The remedy is to recirculate the water and run at higher capacity.

**Water Softening for Can Ice Making.**[9]—"The requisites for first-quality ice are clearness, firmness and freedom from discoloration. These qualities are possessed by ice made from dissolved solids and gases, such as the reboiled distilled water which has, until comparatively recently, been almost exclusively used in the artificial ice industry. Ice frozen from impure water is opaque, discolored, or brittle, depending on the nature of the impurities.

"Freezing water is, in many respects, much like boiling and evaporating it, in that by far the greatest part of the substances dissolved in the water freeze out in the ice made from it. The most effective purification of raw water for ice making is, therefore, that which reduces the objectionable impurities in the water to a minimum. It is now generally recognized that the most effective purification is accomplished by lime-soda softening, followed by sand filtration.

"In the process of the manufacture of ice from raw water, cans of the water are surrounded by the sodium or calcium chloride brine having a temperature of 12 to 18 deg. F. Air under either high pressure (15 to 25 lb.) or low pressure (3 to 5 lb.) is bubbled through the water as it freezes, the high-pressure air being in general more effective. The first ice formed around the sides of the can is usually relatively pure. The dissolved gaseous and solid impurities in the water are frozen and begin to deposit on the face of the ice; but the currents of water set up by the air agitation wash these impurities off the surface of the ice and carry them towards the center of the can. The impurities in the raw water thus become concentrated in the unfrozen water in the middle of the can. If these impurities are insoluble, their accumulation in this unfrozen water usually becomes so heavy that eventually the currents of water set up by the air agitation are not powerful enough to keep the particles in suspension. As a result these white or colored particles are deposited in the ice before the cake is frozen solid, or if the impurities are soluble, as, for instance, sodium salts, their concentration may become so great that freezing is materially retarded. In either case this concentrated impure water, or 'core water' as it is termed, is generally removed, usually with a suction pump, and is replaced with fresh water. The solids and gases left in the core water, or introduced in the fresh

[9] A. S. Behrman. Journal of Ind. and Eng. Chemistry, March, 1921.

water replacing it, appear as white or colored deposits, and as air needles in the core of the ice when the cake is frozen solid.

"The most common, and at the same time the most undesirable class of calcium and magnesium compounds are those causing temporary hardness—that is, the bicarbonates. Just as heating a water of this nature causes precipitation of the normal carbonates, so will freezing it drive off the loosely held 'half-bound' carbon dioxide and cause a precipitation in the ice of the normal carbonates, and possibly of magnesium hydrate. With air agitation these precipitated compounds will be carried more or less completely to the center of the can. Here they will accumulate until it becomes necessary to pump out the heavily laden water and replace it with fresh water. Frequently this accumulation takes place so rapidly that two, and sometimes even three, core pumpings are required. Even with good air agitation, however, the removal of the precipitated compounds to the middle of the can is often not complete, and milky white dots, bubbles and patches are found distributed throughout the clear portion of the ice. Frequently also a white opaque shell of the precipitated carbonates will be found around the lower portion of the cake, where freezing is most rapid. When ice containing the precipitated carbonates melts it leaves this objectionable sediment.

"Softening with lime removes the bicarbonates effectively and cheaply, and leaves in the treated water no products of the reaction beyond the 2.5 to 4 grains per gallon of calcium carbonate and magnesium hydrate generally considered the limit of the lime reaction in the cold.

"The removal of permanent hardness is far less important for ice making than of temporary hardness. Investigations now under way indicate that in a great many cases, possibly all permanent hardness need not be removed, provided that the magnesium which always tends to make white ice is removed from such compounds and is replaced with calcium. This is accomplished of course in the treatment with lime. The calcium sulphate and chloride left in the water as a result of the lime treatment appear to be no more detrimental—and in some cases even less so—than the sodium salts which would result in the removal of the permanent hardness with soda ash. In practically every case ice made from the plain lime treatment is equally as good as or better than when soda is used. In addition, sedimentation in the softening tanks is more complete, thereby reducing the load on the filters. Further, when soda ash is used the carbonate ions in the treated water are lessened and as the ice freezes a much greater concentration in the unfrozen water is required before the ion-product constant is exceeded. As a result the unfrozen water remains clear much longer, and free from particles of the precipitated carbonate that would tend to deposit on the ice, consequently core pumping can be delayed, a less amount of water can be used and the refrigeration thus otherwise wasted, can be saved.

"As little as 0.2 parts per million of iron may cause 'red ice,' that is ice colored reddish brown, principally in the core. Silica and alumina are deposited in the core of the ice cake, imparting a muddy appearance to it, and when this ice melts there remains a slimy sediment. Organic matter is frequently found in objectionable quantity in surface waters, particularly in warm weather. It usually colors the core of the ice a muddy or bright yellow, which is sometimes so objectionable that the ice is saleable with difficulty although it is of good quality in other respects. Lime-soda softening of the raw water followed with sand filtration aided by the use of a coagulant effectively removes the iron, reduces the silica and the alumina usually by from one half to three quarters and greatly lessens the amount of objectionable organic matter. Sometimes bleaching powder can be used to advantage

to remove organic matter. The chief objection to sodium salts (and potassium) is that they accumulate in the core water and retard freezing and are finally deposited as white solids in the ice. If a considerable amount of sodium salts is present the lengthening of the freezing period may be so serious as to require several core pumpings and refillings with fresh water. In addition to this general objection certain sodium compounds have specific undesirable results. Sodium bicarbonate in considerable amount tends to cause brittleness and cracking. Large quantities of sodium sulphate tend towards the formation of a white shell on the outside of the cake, giving the whole cake the appearance of marble even though the interior portion is quite clear. Treatment with lime converts the bicarbonate of soda to the normal carbonate and decreases somewhat the tendency towards brittleness. Softening has no other beneficial effects, as the only way to remove sodium salts is to apply distillation.

"**Checking and Cracking.**—There is a tendency in ice making for the ice to crack and shatter, especially when a low temperature brine is used, and the only explanation that can be offered is that the ice is frozen under some internal strain. It is quite possible that the presence of bicarbonates is chiefly responsible for this strain. During the freezing process, while the half-bound carbon dioxide is trying to escape, the ice continues to crystallize, entrapping bubbles of gas and particles of the precipitated compound which are readily visible. The ice thus formed is comparable to a metal casting full of blow holes and impurities and is in consequence inherently weak and brittle.

"Some weight is given this hypothesis by the general experience that removing the bicarbonates of calcium and magnesium from the water by treatment with lime results in the production of much clearer and firmer ice, and frequently permits the use of lower brine temperatures. Further in a recent series of experiments ice was frozen from water to which had been added varying amounts of sodium bicarbonate. In all cases except the lowest concentration (10 grains per gallon) the ice formed was quite brittle, cracked readily, and showed evidence of a bubbly structure. Analysis of the melted core ice showed the conversion of the bicarbonates to the normal carbonate in all cases to the extent that the normal carbonate alkalinity averaged 35 per cent of the bicarbonate alkalinity.

"Finally the question arises as to the limiting quantities of the various impurities that a raw water can carry, and still make first class ice. This is not known in all cases. Obviously in the cases of the bicarbonates of calcium magnesium and iron, the limiting concentrations are their own solubilities, since softening with lime leaves the same residual content regardless of the original concentration. It is also probable that the permissible maximum of silica and alumina is not exceeded in natural waters if treated with lime.

"With regard to sodium salts, and to calcium sulphate and chloride, investigations tend to indicate that when the total soluble salt content of a raw water exceeds 30 to 40 grains per gallon exclusive of the temporary hardness, first quality raw water ice cannot be made even with water softening high-pressure air agitation."

**Checking and Cracking of Ice.**—The advent of raw water ice making has brought many problems, and one of these is that of cracking and checking of the ice cake during freezing, or during the pulling period or even up to the period of storing in the daily ice storage room. The

causes of checking are many, and a few of the principal ones will be given in the following:[9a]

1. Poor brine circulation in the brine tank.
2. Allowing the cans to temper while standing in draughts of warm fresh air, especially if the sprinkler system is used, though tempering will be found to be good for the ice in some cases if the cake is not subject to draughts.
3. Filling the cans with water that is too warm.
4. Rough handling of cans by tank men during core sucking, hoisting and dumping.
5. Lowering cans into the thawing tank too suddenly, or using a thawing water of too high a temperature.
6. Leaky cans.
7. Too low a brine temperature (approximately 44 hr. is the correct freezing time for 300 lb. cans).
8. Presence of bulged cans.
9. Brine level below the level of the water in the cans.
10. Dissolved calcium carbonate in the water to be frozen, especially when calcium sulphate and sodium sulphate are present.

From the preceding list it is clear that the causes of checking are not clear and definite, and therefore the correction of this trouble is one of the most difficult of any that the refrigerating engineer may be called upon to solve. However, it is positive that rough handling, dropping of the core sucking rod into the ice cavity and pounding and pumping of the cans are very bad for the production of sound ice. Sometimes the lowering of the cans quickly into comparatively hot water is better than to do it slowly in cold water, and again, in other cases, the use of 60 deg. F. water in the dip tank and a lowering rate of the can such as would require $7\frac{1}{2}$ minutes to submerge the can has been found best. The old form of sprinkler device at the dump is thought to be more likely to cause checking than the dip tank, whereas the use of air agitation of the dip tank is considered to give better results than the tank without agitation. The accepted reason for can ice cracking is because of freezing stresses due to the expansion and contraction of the water and ice during the freezing process. Some ice is cracked before it is removed from the cans.

**Piping.**—The method of cooling brine in the ice tank for a long time has been to use direct expansion piping in the brine tank (Fig. 263).

[9a] E. S. Ormsby, National Association of Practical Refrigerating Engineers, 1923.

These pipes are arranged to run the long dimension of the tank, and are placed in stands between the individual ice cans, thereby permitting

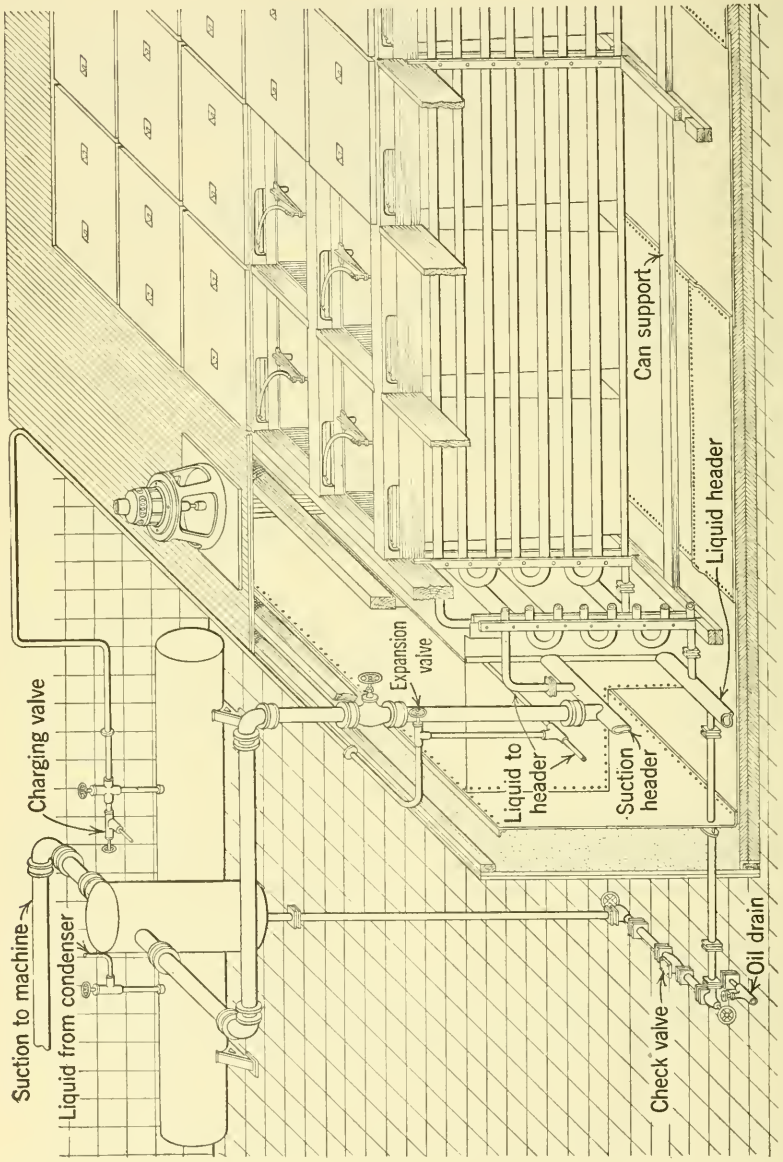



Fig. 263.—Can Ice Making—Tank Assembly.

close proximity between the cans and the heat absorbing medium. These pipe coils (usually $1\frac{1}{4}$-in. pipe) with the old system of operation

were not found to be very effective because they had to be so operated as to prevent liquid returning to the compressor, and the flooded system has entirely superseded it. The flooded system (shown in Fig. 264) is not a device for securing more refrigeration per pound of ammonia but only for providing efficient heat transfer in the evaporating surfaces. Piping which was originally calculated for 12.5 or 15 B.t.u. for the value of the coefficient of heat transfer can transmit twice that amount

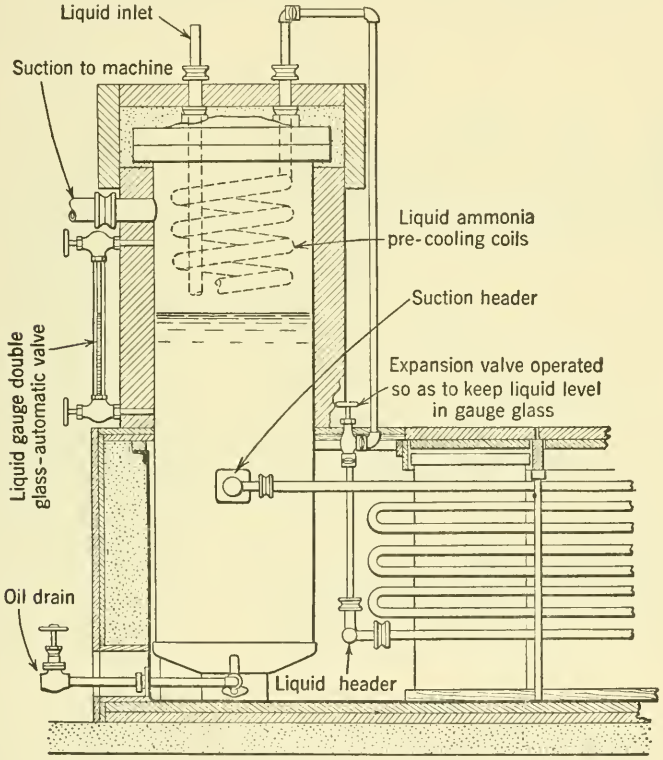

Liquid inlet
Suction to machine
Liquid ammonia pre-cooling coils
Suction header
Liquid gauge double glass-automatic valve
Expansion valve operated so as to keep liquid level in gauge glass
Oil drain
Liquid header

Fig. 264.—The Accumulator and Piping in Ice Tank.

with the flooded system with the same amount of agitation. As piping has been considered to be the cheapest part of the plant it has been the custom to specify 250 to 300 linear ft. of $1\frac{1}{4}$-in. pipe per ton of ice making. For a 100-ton ice plant there would be required under these circumstances from 5 to 6 miles of pipe which must be made very tight unless a loss of ammonia from the system and the formation of ammonia chloride is not to be faced during the ice making season without any means of repair except a complete shut-down and the removal of the

brine from the tank. In the old ice tank designs the process of filling and replacing the cans was very likely to injure the piping.

The alternative arrangement, which is becoming popular in certain sections, is to use the shell and tube brine cooler, submerged in the brine, and arranged for a single pass of the brine. The ammonia, instead of boiling out of a 1500-ft. length of pipe, has a distance of 8 to 10 ft. to travel from the expansion valve to the beginning of the suction return line to the compressor (Fig. 171). The objections to the use of the shell and tube brine cooler have been a fear that the cooler could not be made tight, especially where the tubes were expanded into the tube sheet, and that as the cooling surfaces were only at one end (or both ends) of the tank, the brine temperature would be non-uniform in the tank. There is also a feeling that the resistance to the flow of the brine through the cooler is too great, requiring thereby too much expenditure of power to provide circulation of the brine around the tank and the cans. Apparently some of these fears are not justified as a brine temperature variation of not more than $\frac{1}{2}$ deg. F. is usually found if good agitation is provided, and at the present time there seems to be no difficulty in being able to make both brine coolers and condensers of the shell and tube construction and to keep the tubes tight in the tube sheet by the simple expedient of expanding into the sheet. The shell and tube cooler for this work is shown in Fig. 172, which is designed for a single pass. The usual velocity of the brine in the tubes is 400 ft. per min., and the coefficient of heat transfer under these conditions is 90 B.t.u. per hour. The objection that valuable space will be taken up by the cooler is answered in a way by the statement that as the piping

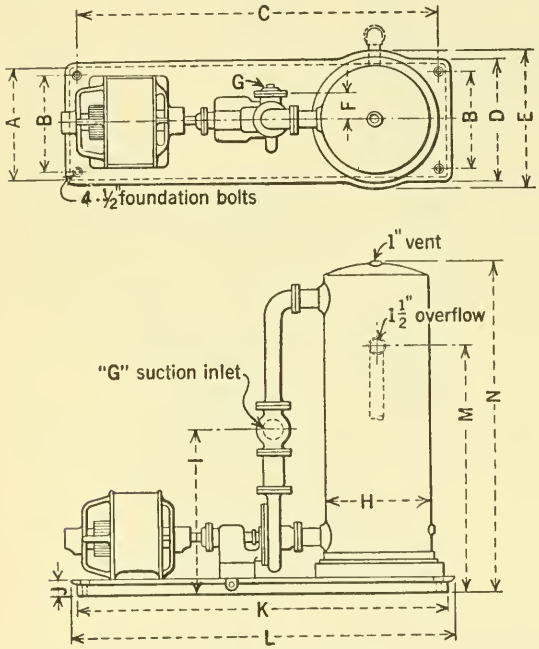



Fig. 265.—Core Sucker.

is not placed between the cans, these can be placed on closer centers. This is especially true in the case of cans in baskets which permit a lighter tank frame construction. The shell brine cooler lends itself to the flooded operation, and by means of the separator and drain (Fig. 170) back to the cooler the danger due to a "slop over" to the compressor is removed.

**Precooling Water for the Cans.**—If the water goes to the cans at 70 deg. F., then 38 B.t.u. must be removed per pound before freezing can begin, and this action of cooling can be performed more advantageously in some other device, as, for example, in a special water

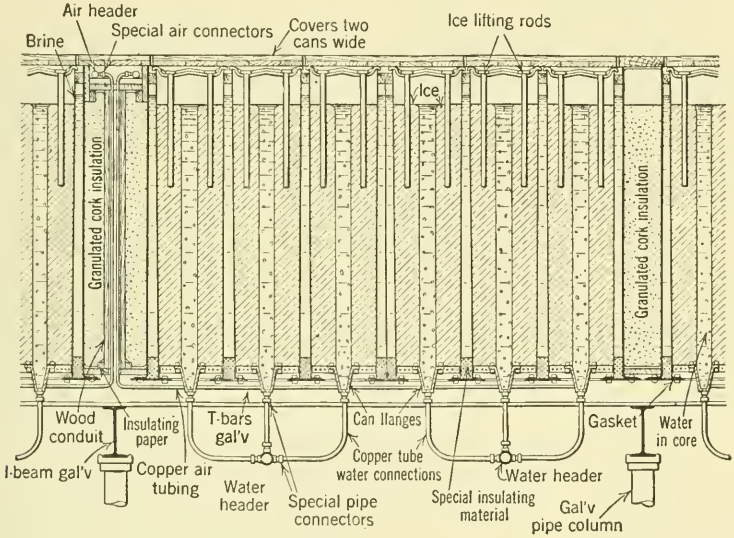



Fig. 266.—The Arctic-Pownall System.

cooler. It should be kept in mind, however, that the use of a water cooler may not give good results unless operating care is observed. It is easy to lose all the advantage of precooling. Tests indicate that the water in the cans, if these are filled at 70 deg. F., can be lowered to 32 degrees in about 15 to 20 minutes. In the Arctic-Pownall process (Fig. 266) the precooling tank is carefully insulated and only slight loss will be occasioned on account of heat leakage from the air to the water.

**Freezing Processes.**—A number of designs used in the manufacture of ice a few years ago are spoken of but rarely now. The *Hume* froze a cake of four times the usual size (22 in. by 44 in.) which was sawed into standard sized cakes so as best to eliminate the core. Core sucking and the filling of the cavity with distilled water was spoken of as the

*Ulrich* process. The *Bishop* process, a modification of the plate in order to fit the conditions of the can system, was arranged so that the water entered the can at the bottom and overflowed at the top. As the can was kept cool in the usual manner with brine, freezing took place from all four sides and a fairly clear cake of ice resulted. The can was arranged so as to be stationary and the ice had to be loosened in the can by circulating warm brine and the cakes lifted by means of hooks frozen into the ice for that purpose. The *Quick-freeze* system attempted to cut down the time of freezing by the use of a special pipe arrangement

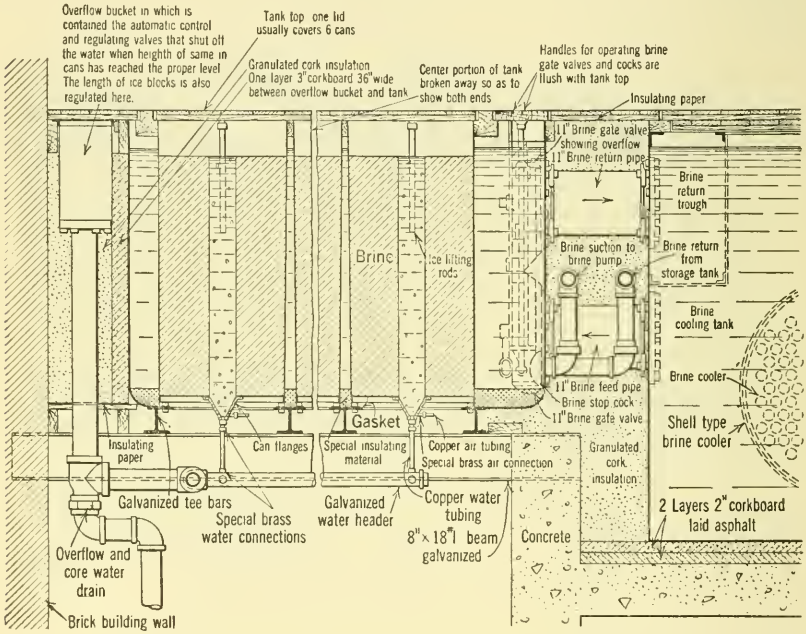

Overflow bucket in which is contained the automatic control and regulating valves that shut off the water when heighth of same in cans has reached the proper level The length of ice blocks is also regulated here.
Tank top one lid usually covers 6 cans
Granulated cork insulation One layer 3"corkboard 36"wide between overflow bucket and tank
Center portion of tank broken away so as to show both ends
Handles for operating brine gate valves and cocks are flush with tank top
Insulating paper
11" Brine gate valve showing overflow
11" Brine return pipe
Brine return trough
Brine
Ice lifting rods
Brine suction to brine pump
Brine return from storage tank
Brine cooling tank
Brine cooler
11" Brine feed pipe
Brine stop cock
11" Brine gate valve
Shell type brine cooler
Gasket
Insulating paper
Can flanges
Special insulating material
Copper air tubing
Special brass air connection
Granulated cork insulation
Galvanized tee bars
Special brass water connections
Galvanized water header
Copper water tubing
8"×18"I beam galvanized
Concrete
2 Layers 2" corkboard laid asphalt
Overflow and core water drain
Brick building wall

Fig. 267.—The Arctic-Pownall System.

projecting upward from the bottom of the can and designed to circulate brine. As heat was absorbed by each pipe as well as by the usual process, the time of freezing of a standard cake of ice would be reduced. In the plate system there was the *Hill-Ray* method, the only apparent advantage of which was an arrangement of wedges by which the size of the cake was reduced and the amount of sawing diminished in proportion. The *Center-Freeze* method froze the cake from two sides at the same time, the pipes being spaced about 5 to 6 in. apart, and it was claimed that a reduced time of freezing was accomplished. The finished cake would of course have holes in it where the pipes had been during freezing.

The *Holden* Regulation system is still spoken of as a present-day possibility. In this a hollow drum, kept cold by the use of brine, or direct expansion, is made to revolve with its surface in contact with a trough of water (see Lard Cooling, Chapter XIV). The ice film is scraped off and the snow is compressed hydraulically to the size of cake desired. This method is very nearly ideal, the cost per ton is low, and the process is continuous. Unfortunately there is no known means of producing thereby a crystal cake, and this method has not been used commercially for years. Finally, there is the *Jewell* process, a modification of which (the Arctic-Pownall) is a most successful present-day design.

The Jewell process uses a double-walled can which is stationary. The space between the two walls is filled with brine, the brine entering at the top and leaving at the bottom. The bottom of the can is insulated and contains the air connections for agitation. The Arctic-Pownall arrangement is shown in Figs. 266 to 268. The brine is cooled by means of shell and tube brine coolers, and the agitation is vigorous. The ice tank needs to be open at the bottom, and is supported on I beams, as shown. The filling of the cans, dropping, and the refilling of the cores are performed by the means of piping connected to the can flanges. A number of 300- or 400-lb. cans—from 28 to 84—are connected so as to form an ice tank. These cans are all interconnected so that can filling, core dropping, etc., can take place in unison by the operation of one valve. By this method all cans are filled at the same level which can be adjusted, and the water for can filling can be stored in a tank to the correct amount (previously cooled if desired).

The usual method of operation is as follows: The plant is made up of a number of tanks all of similar construction. The water for the cans of one of these tanks has been precooled by an arrangement shown in Fig. 268, and by the operation of one valve the tank is emptied and the cans in one of the tanks are filled. The air agitation is started in all the cans of the tank by the opening of one valve and the brine is circulated about the cans by the operation of two quick-acting valves. The advantages of this system are in the much reduced labor costs, as one man is claimed to be able to pull at the rate of 175 tons per 24 hours, no mechanical injury to the cans, no brine on the covers of the tank, economy in the use of brine in the precooling of the water for the cans and by the use of this heated brine for the thawing of the ice out of the cans, the use of low-pressure air for agitation entirely and no chance of white (marble) ice due to shut-downs of power. The first cost is considerably greater (10 to 15 per cent) than the usual medium-pressure air can system, and the design calls for vigorous brine agitation. However, it is possible to operate with less loss of refrigeration, with less wear

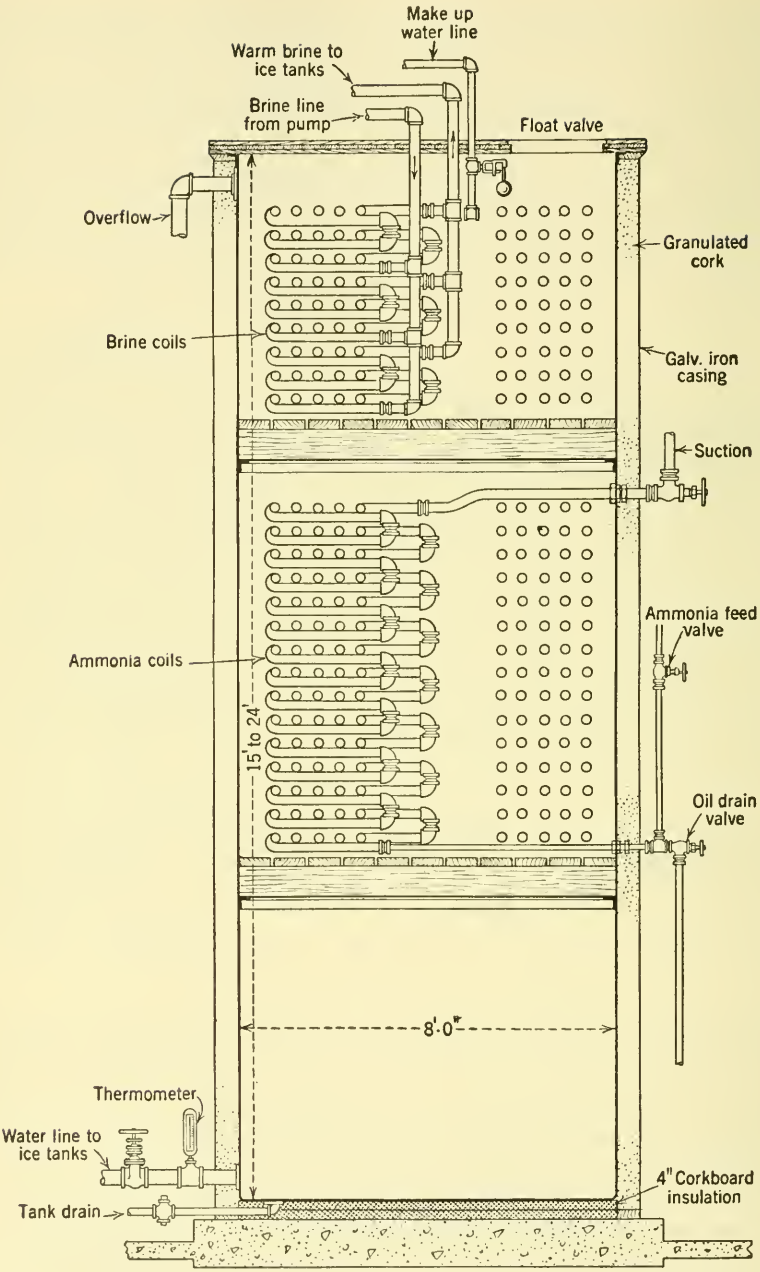



FIG. 268.—Water Cooler and Tank.

and tear on the equipment, and with less labor on the tank room floor with this system than with any other.

**The Design of an Ice Making Plant.**—Ice plant design is different in various parts of the United States and even in different parts of the same city. In order that design details may be emphasized, some material already treated in this chapter will be repeated as far as it affects the matter of design.

TABLE 91

FREEZING TANK FRAMEWORK AND COVERS FOR STANDARD CANS

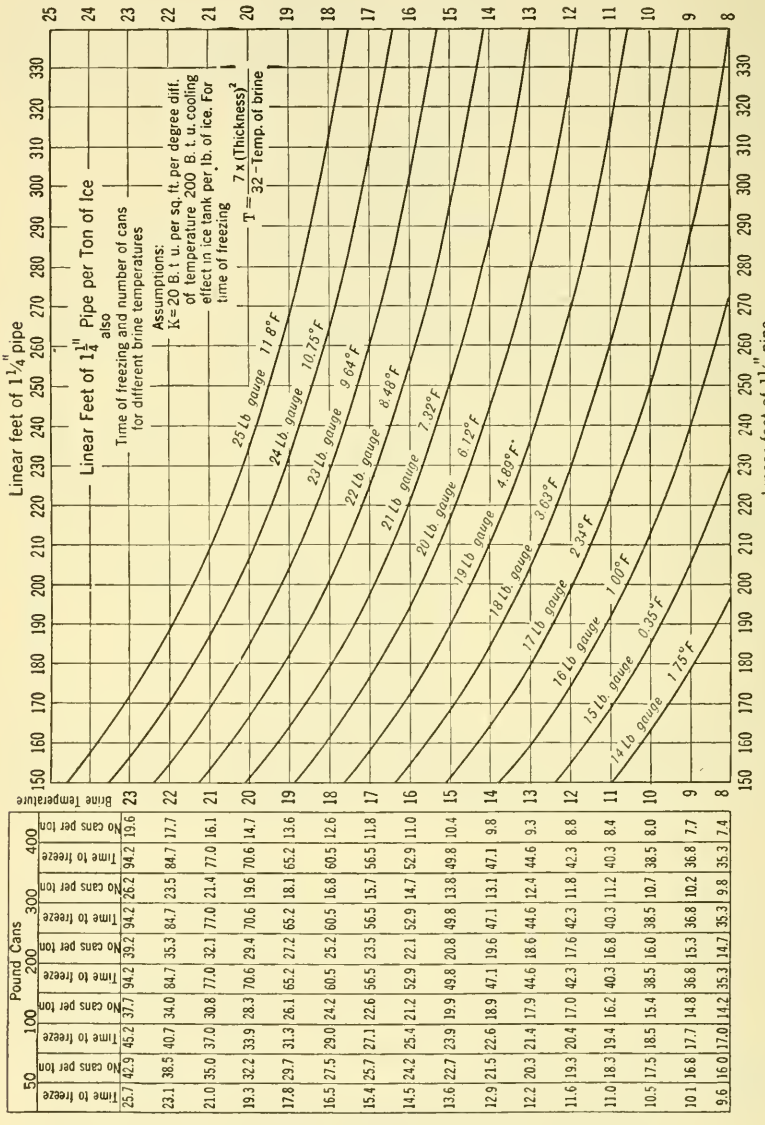
| Weight of Cake of Ice, Pounds | Can Dimensions Measured at Top, Inches | | Framework Dimensions, Inches, Center to Center | | Covers, Weight per Can, Pounds | |
|---|---|---|---|---|---|---|
| | Width | Length | Width | Length | Distilled or opaque | Raw water |
| 25 | 4 | 9 | 7 | 11 | 28 | 28 |
| 50 | 5 | 12 | 8 | 14 | 30 | 30 |
| 60 | 5 | 14 | 8 | 16 | 30 | 30 |
| 100 | 8 | 16 | 11 | 18 | 32 | 32 |
| 200 | $11\frac{1}{2}$ | $22\frac{1}{2}$ | $14\frac{1}{2}$ | $24\frac{1}{2}$ | 36 | 36 |
| 300 | $11\frac{1}{2}$ | $22\frac{1}{2}$ | $14\frac{1}{2}$ | $24\frac{1}{2}$ | 36 | 36 |
| 400 | $11\frac{1}{2}$ | $22\frac{1}{2}$ | $14\frac{1}{2}$ | $24\frac{1}{2}$ | 36 | 36 |

NOTE.—Covers and frames are made of finished lumber. Covers are made of two thicknesses of $1\frac{1}{4}$-in. material screwed together with wood screws and have recessed opening with pin for lifting hook. Unless otherwise specified material will be oak for all standard equipment.

The present tendency is towards the reduction of the cost of labor to the minimum. This means the use in the larger plants of the 400-lb. cans instead of the 300-lb. cans, as the time of freezing is the same for both, and—with the exception of certain patented tank designs—of the use of medium or high-pressure air agitation. In fact, it has been pointed out that certain raw waters heavy in certain salts cannot make the best ice with low-pressure air agitation.

The tendency of the times is also for the lifting of gangs of ice cans at a time, say from five to eighteen or more, though the general practice is not more than six. The advantage of lifting more than two cans at a time depends on local conditions, and the entire subject is one for careful analsysis if the most economical arrangement is to be made. The heavy

agitation is for the purpose of increasing the coefficient of heat transfer of the pipes in the tank from 20 B.t.u. (for flooded operation) per square


Linear feet of 1¼" pipe
Linear Feet of 1¼" Pipe per Ton of Ice
also
Time of freezing and number of cans
for different brine temperatures
Assumptions:
K = 20 B. t. u. per sq. ft. per degree diff.
of temperature 200 B. t. u. cooling
effect in ice tank per lb. of ice. For
time of freezing
T = 32 − Temp. of brine
7 x (Thickness)²
25 Lb. gauge 11.6°F
24 Lb. gauge 10.73°F
23 Lb. gauge 9.64°F
22 Lb. gauge 8.48°F
21 Lb. gauge 7.32°F
20 Lb. gauge 6.12°F
19 Lb. gauge 4.89°F
18 Lb. gauge 3.63°F
17 Lb. gauge 2.34°F
16 Lb. gauge 1.00°F
15 Lb. gauge 0.35°F
14 Lb. gauge 1.75°F
Brine Temperature
Linear feet of 1¼" pipe

| Brine Temperature | Pounds Cans 50 | | 100 | | 200 | | 300 | | 400 | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Time to freeze | No cans per ton | Time to freeze | No cans per ton | Time to freeze | No cans per ton | Time to freeze | No cans per ton | Time to freeze | No cans per ton |
| 23 | 25.7 | 42.9 | 45.2 | 37.7 | 94.2 | 39.2 | 94.2 | 26.2 | 94.2 | 19.6 |
| 22 | 23.1 | 38.5 | 40.7 | 34.0 | 84.7 | 35.3 | 84.7 | 23.5 | 84.7 | 17.7 |
| 21 | 21.0 | 35.0 | 37.0 | 30.8 | 77.0 | 32.1 | 77.0 | 21.4 | 77.0 | 16.1 |
| 20 | 19.3 | 32.2 | 33.9 | 28.3 | 70.6 | 29.4 | 70.6 | 19.6 | 70.6 | 14.7 |
| 19 | 17.8 | 29.7 | 31.3 | 26.1 | 65.2 | 27.2 | 65.2 | 18.1 | 65.2 | 13.6 |
| 18 | 16.5 | 27.5 | 29.0 | 24.2 | 60.5 | 25.2 | 60.5 | 16.8 | 60.5 | 12.6 |
| 17 | 15.4 | 25.7 | 27.1 | 22.6 | 56.5 | 23.5 | 56.5 | 15.7 | 56.5 | 11.8 |
| 16 | 14.5 | 24.2 | 25.4 | 21.2 | 52.9 | 22.1 | 52.9 | 14.7 | 52.9 | 11.0 |
| 15 | 13.6 | 22.7 | 23.9 | 19.9 | 49.8 | 20.8 | 49.8 | 13.8 | 49.8 | 10.4 |
| 14 | 12.9 | 21.5 | 22.6 | 18.9 | 47.1 | 19.6 | 47.1 | 13.1 | 47.1 | 9.8 |
| 13 | 12.2 | 20.3 | 21.4 | 17.9 | 44.6 | 18.6 | 44.6 | 12.4 | 44.6 | 9.3 |
| 12 | 11.6 | 19.3 | 20.4 | 17.0 | 42.3 | 17.6 | 42.3 | 11.8 | 42.3 | 8.8 |
| 11 | 11.0 | 18.3 | 19.4 | 16.2 | 40.3 | 16.8 | 40.3 | 11.2 | 40.3 | 8.4 |
| 10 | 10.5 | 17.5 | 18.5 | 15.4 | 38.5 | 16.0 | 38.5 | 10.7 | 38.5 | 8.0 |
| 9 | 10.1 | 16.8 | 17.7 | 14.8 | 36.8 | 15.3 | 36.8 | 10.2 | 36.8 | 7.7 |
| 8 | 9.6 | 16.0 | 17.0 | 14.2 | 35.3 | 14.7 | 35.3 | 9.8 | 35.3 | 7.4 |

FIG. 269.—Ice Freezing Diagram.

foot per one degree difference per hour (the value $k$) to 25 or possibly to 30. The particular gain here is to permit the amount of piping in the tank to be reduced, and also to reduce the time of freezing of the water

TABLE 92

SIMPLE BELTED AIR COMPRESSORS FOR MEDIUM-PRESSURE RAW WATER PLANTS

| Number of Cans | Tons of Ice | Cubic Feet of Air per Minute | Size, Inches | | R.p.m. Required | Horsepower Motor Required | Maximum Compressed Speed, R.p.m. | Maximum Pressure for Comp., Pounds | Pulley, Inches | | Pipe Connections, Inches | | | Approximate Floor Space, Inches | Weight Boxed |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Diameter | Stroke | | | | | Diameter | Face | Suction | Discharge | Water | | |
| 70 | 5 | 14.0 | 4 | 5 | 250 | 3 | 350 | 125 | 30 | 4½ | 1¼ | 1¼ | ½ | 55×22 | 850 |
| 84 | 6 | 16.8 | 5 | 5 | 225 | 3 | 350 | 125 | 30 | 4½ | 1¼ | 1¼ | ½ | 55×22 | 850 |
| 112 | 8 | 22.4 | 6 | 5 | 200 | 5 | 350 | 100 | 30 | 4½ | 1½ | 1½ | ½ | 57×22 | 950 |
| 144 | 10 | 28.8 | 7 | 5 | 205 | 5 | 350 | 50 | 30 | 4½ | 2 | 2 | ½ | 59×22 | 1060 |
| 168 | 12 | 33.6 | 7 | 5 | 215 | 5 | 350 | 50 | 30 | 4½ | 2 | 2 | ½ | 59×22 | 1060 |
| 210 | 15 | 42.0 | 7 | 5 | 225 | 7½ | 350 | 50 | 30 | 4½ | 2 | 2 | ½ | 59×22 | 1060 |
| 280 | 20 | 56.0 | 7 | 6 | 240 | 10 | 350 | 100 | 36 | 5½ | 2 | 2 | ½ | 60×26 | 1305 |
| 360 | 25 | 72.0 | 8 | 6 | 250 | 10 | 350 | 50 | 36 | 5½ | 2½ | 2½ | ½ | 68×26 | 1475 |
| 420 | 30 | 84.0 | 9 | 6 | 230 | 15 | 325 | 25 | 36 | 5½ | 3 | 3 | ¾ | 70×26 | 1550 |
| 484 | 35 | 96.8 | 9 | 8 | 230 | 15 | 325 | 100 | 42 | 8½ | 3 | 3 | ¾ | 81×34 | 2610 |
| 560 | 40 | 112.0 | 10 | 8 | 210 | 20 | 325 | 50 | 42 | 8½ | 3½ | 3½ | ¾ | 85×34 | 2800 |
| 720 | 50 | 144.0 | 10 | 8 | 235 | 20 | 325 | 50 | 42 | 8½ | 3½ | 3½ | ¾ | 85×34 | 2800 |
| 840 | 60 | 168.0 | 12 | 8 | 220 | 25 | 325 | 30 | 42 | 8½ | 4 | 4 | 1 | 87×34 | 2925 |
| 1120 | 80 | 224.0 | 14 | 9 | 200 | 30 | 300 | 25 | 54 | 9½ | 5 | 5 | 1¼ | 94×39 | 3800 |
| 1440 | 100 | 288.0 | 14 | 9 | 210 | 30 | 300 | 25 | 54 | 9½ | 5 | 5 | 1¼ | 94×39 | 3800 |
| 1680 | 120 | 336.0 | 16 | 10 | 200 | 40 | 275 | 35 | 60 | 10½ | 6 | 6 | 1¼ | 107×46 | 6000 |
| 2160 | 150 | 432.0 | 16 | 10 | 215 | 50 | 275 | 35 | 60 | 10½ | 6 | 6 | 1¼ | 107×46 | 6000 |
| 2880 | 200 | 576.0 | 14 | 9 | 210 | 30 | 300 | 25 | 54 | 9½ | 5 | 2 | 1¼ | 94×39 | 7600 |
| | | | 14 | 9 | 210 | 30 | 300 | 25 | 54 | 9½ | 5 | 5 | 1¼ | 94×39 | |

Based on 14 cans per ton. For less figure .2 cu. ft. air per minute and use proper compressor, working pressure of compressors 25 lb. Based on belted, enclosed type, horizontal, D. A. compressor with unloader and foundation bolts. No mechanically operated valves included.

in the cans, although the empirical formula for the time of freezing has no term to denote the effect of agitation on the freezing time. The latest practice is to raise the evaporating or suction pressure as high as possible, say, from 22 to 25 lb. gage. This means a high boiling temperature of the ammonia and in consequence greater capacity for the compressor and less cost in horse power per ton of refrigeration, but this of course increases the auxiliary horse power. The relation of number of cans, brine temperature and pipe surface is well shown in Fig. 269 and Table 92.

The matter of the calculation of an ice plant problem may, then, be different in certain details, but the manner of the calculation can be shown clearly by the problem, the details of which would need to be carefully decided for the individual case by an analysis of the costs, by a consideration of the power and labor items, as follows: The ice plant is to be a 100-ton raw water can ice plant, for 24-lb. suction and 154.5-lb. condenser pressure, using 70 deg. F. water to the cans and 16 deg. brine. The outside air temperature is 90 degrees, and the temperature of the earth is 50 deg. F. Use high pressure air agitation, 30 lb. at the compressor and 15 lb. at the cans. Two dehumidifiers, brine cooled, to cool 200 cu. ft. of free air from 90 to 20 deg. F. per minute.

Two tanks[10] of $\frac{1}{4}$-in. steel will be used, 5 ft. 1 in. high, each to hold 600 400-lb. cans 58 in. high. Each tank will have the one accumulator and two 12-in. brine agitators—using 5 hp. each—with motors direct connected to vertical shafts. Can centers are to be 2 ft. $\frac{3}{8}$ in. by $14\frac{1}{4}$ in., and the tanks are to be insulated with 12 in. of granulated cork on the sides and with 5 in. of cork board on the bottom. Lift five cans at a time, and use $1\frac{1}{4}$-in. full weight pipe.

**1st. To Find the Heat Removed per Pound of Ice Placed in Daily Storage.**

| | | |
|---|---|---|
| To cool the water from 70 to 32 deg. F. per 1 lb. of ice.... | 38.0 | B.t.u. |
| To freeze the water per 1 lb. of ice..... | 144.0 | " |
| To cool the ice from 32 to 16 deg. (spec. heat of ice = 0.5) | 8.0 | " |
| Total..... | 190.0 | B.t.u. |
| Add 10 per cent for non-computable losses..... | 19.0 | |

Heat leakage through the insulation (calculated as in the problem on cold storage)[11]
Sides, 12-in granulated cork, 2-$\frac{7}{8}$ in. and g. cypress.

$$U = \frac{1}{\frac{1}{1.4} + \frac{2}{1} + \frac{12}{0.296}} = 0.0231.$$

[10] These details follow standard practice.

[11] See Chapter XIV.

Bottom, 5-in. corkboard, 9-in. concrete.

$$U = \frac{1}{\frac{5}{0.308} + \frac{9}{5.3}} = 0.0558.$$

Heat leakage: Sides $1187 \times 0.0231 \times (90 - 16) = 2030$ B.t.u.
Bottom $3420 \times 0.0558 \times (50 - 16) = 6490$ "
Total = 8520 B.t.u.

$$= 4260 \text{ B.t.u. per hour per tank of } = \frac{4260 \times 24}{2000 \times 50} = 1.02 \text{ B.t.u. per lb. of ice.}$$

The air for agitation is cooled after compression first by water and then by brine (using two tanks with a four-way valve for alternate change of direction of flow). The brine in refrigerating the air adds a certain amount to the load on the compressor, as, for example, the heat removed from each pound of the air may be considered as 35.0 B.t.u., and the specific volume of the air as 14.2 cu. ft. Then

$$\frac{200}{14.2} \times 35.0 = 493 \text{ B.t.u. per minute,}$$

or, in terms of the ice made,

$$\frac{493 \times 60 \times 24}{100 \times 2000} = 3.56 \text{ B.t.u. per lb.}$$

The agitation of the brine, as already stated, assists in the heat transfer, and keeps the brine temperature constant. Agitation always tends to heat up the brine, and it is usual to figure that the entire heat equivalent of the pump horse power is converted into heat and gas to be neutralized (refrigerated) by the action of the compressor. In this case the amount becomes

$$\frac{20 \times 42.4 \times 60 \times 24}{100 \times 2000} = 6.1 \text{ B.t.u. per lb. ice.}$$

The total refrigeration per one pound of ice made is then

$$209 + 1.02 + 3.56 + 6.1 = 219.7 \text{ B.t.u.}$$

In round numbers the value $220/144 = 1.53$ is the ratio of the number of tons of refrigeration capacity required per ton of ice made according to the calculation. The value generally used is 1.6, although some engineers go to the extreme by the use of the ratio 2.0, but such a

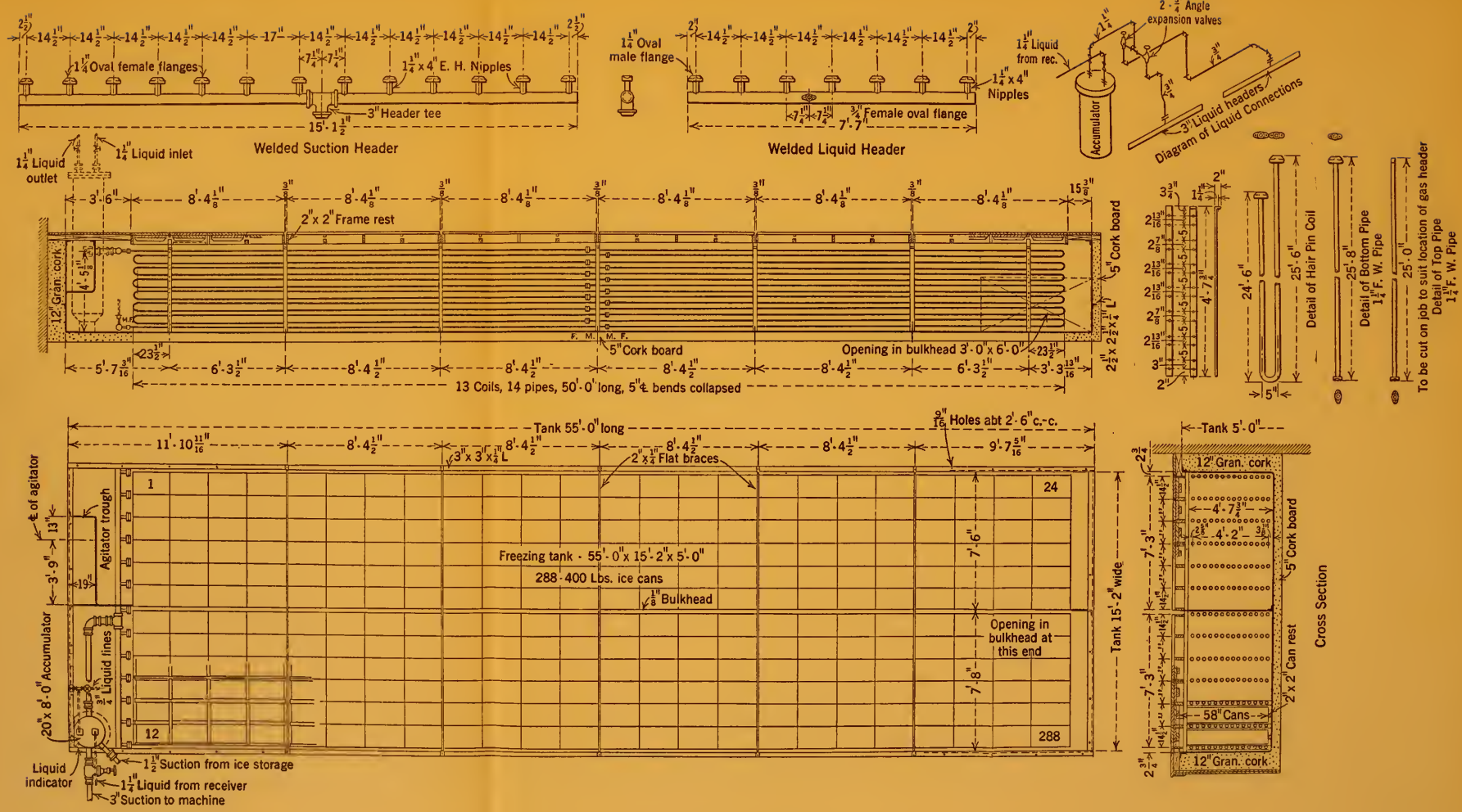



Fig. 270.—Can Ice Tank Design.

*To face page 416.*

to consider first the mechanical efficiency of the equipment; second, the cost of the investment; and third, the plant operation. Medium-sized plants have produced ice at the rate of 37 to 38 kw.-hrs. per ton of ice as the yearly average with a load factor of 60 to 70 per cent, but the cost of power is only one of a number of costs. In all likelihood it is possible that with good agitation and with the flooded system (which makes it possible to decrease the amount of gas in the evaporating system, and by having a heavy feed with some liquid in the coils for the entire length of the direct expansion piping) a value of 30 or more can be obtained for $k$,[12] but a design of this sort requires more care in the operation than does the ordinary plant.

**Selecting the Compressor and the Compressor Drive.**[13]—Selecting the compressor and the compressor drive has become a difficult problem in these days, with the many types and designs which prevail. However, there are a few points of particular prominence which are worth mentioning. It is a mistake to consider that clearance in an ammonia cylinder means an appreciable loss of economy. Clearance affects the capacity of a compressor, but it has not been proved that it increases the horse power per ton of refrigeration to any great extent except for the friction of the machine which remains nearly constant though it is *relatively* larger at small than at full loads, per ton of refrigeration.

The space allotted to the compressor frequently decides the type of compressor. The principle of the design makes one feel that the vertical single-acting compressor using the suction valve in the piston is the correct design in order to reduce the superheating of the suction gas by the cylinder walls to a minimum, but this is also true of the horizontal double-single-acting design. In each case there is a uniflow principle of gas flow. The factor 0.86 for the volumetric efficiency is a term to allow for this, although the figure mentioned may be a little low for the vertical single-acting compressor. However, one is led to believe that the semi-enclosed type of vertical compressor is becoming much less popular than, say, 10 years ago. It has a reciprocating rod type of stuffing-box which is much more difficult to keep tight than the rotary type used in the enclosed compressors and is likely to have more friction in the stuffing-box. In Great Britain the semi-enclosed type is not used at all, and the modern type of horizontal compressor has not been built until recently, while the three- and four-cylinder enclosed machine is very popular and uses forced lubrication throughout. In the United States every manufacturer has a design of a small compressor of the vertical

[12] Thomas Shipley, National Association of Practical Refrigerating Engineers, 1926.

[13] See Chapters II and XIX.

single-acting type, but the majority have horizontal compressors only for the larger sizes. The York Manufacturing Company has developed an enclosed compressor of large capacities, using three cylinders, and going up from 300 to 400 tons of refrigeration capacity.

As regards the drive, for electric-driven compressors[14] requiring 75 hp. and over, a synchronous motor generally should be used with direct connection to the shaft. The gain is in a greater overall efficiency as there are no belt or other transmission losses which in other cases usually amount to from 6 to 8 per cent. This type of motor is particularly good because of its ability to improve the power factor of the circuit as the motor may be made to operate as a synchronous condenser. The disadvantage, which is true also of the squirrel-cage induction and other types of motors, is the impossibility of varying the speed of the compressor. Variation of the capacity can then be obtained only by the judicious use of clearance pockets, a device which has become quite usual. If a steam engine is used, the uniflow engine is the most popular because of the flat water rate curve and the low steam consumption per horse power hour. If the oil engine is decided on there are a number of types to choose from. In any case the power delivered to the shaft should be (see Fig. 31 for 24-lb. suction and 154.5-lb. condenser pressures) 1.10 hp. per ton of refrigeration.[14a] Each of the two compressors will have a capacity of $76.6 \times 1.1 = 84.3$ hp. at the shaft. The piston displacement may be found from the same set

[14] See Chapter XXIII.

[14a] The capacity and the horse power per ton (of refrigeration) can be calculated as follows, for 24.0 lb. gage suction and 154.5 lb. gage condenser pressures:

$$i''_{24\ \text{lb. g.}} - i'_{86\ \text{deg. F.}} = 614.9 - 138.9 = 476\ \text{B.t.u.}$$

= net refrigeration per 1 lb. liquid ammonia.

Theoretical volume boiled off per ton of refrigeration per minute $= \frac{200}{476.0} \times 7.28$ = 3.06 cu. ft., and with a volumetric efficiency of 0.86 the piston displacement per ton per minute becomes 3.06/0.86 = 3.56 cu. ft., not allowing for clearance. (See Chapter II.)

The horse power per ton of refrigeration is obtained from

$$W = \frac{n}{n-1} p_1 v_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]$$

$$= \frac{1.28}{0.28} \times 144 \times 38.7 \times 3.56 \left[1 - \left(\frac{169.2}{38.7}\right)^{\frac{0.28}{1.28}}\right]$$

$$= 34{,}500\ \text{ft.-lb.} = 1.05\ \text{hp.}$$

of curves, and for the conditions stated this becomes 3.6 cu. ft. per ton of refrigeration per minute. Therefore, a 76.6-ton machine will require $76.6 \times 3.6 = 276$ cu. ft. per min. for each of the compressors. The type of the compressor chosen will affect the rotative speed, but the present practice is to use some form of light weight valve of the plate or the ribbon type. The heavy poppet valve is practically obsolete. Should a synchronous motor be used clearance pockets may be employed to advantage in regulating the capacity of the compressor.

Ice-making plants have use for *two suction pressures*,[15] the second being on the fore-cooler at, say, 35-lb. gage. This is especially true

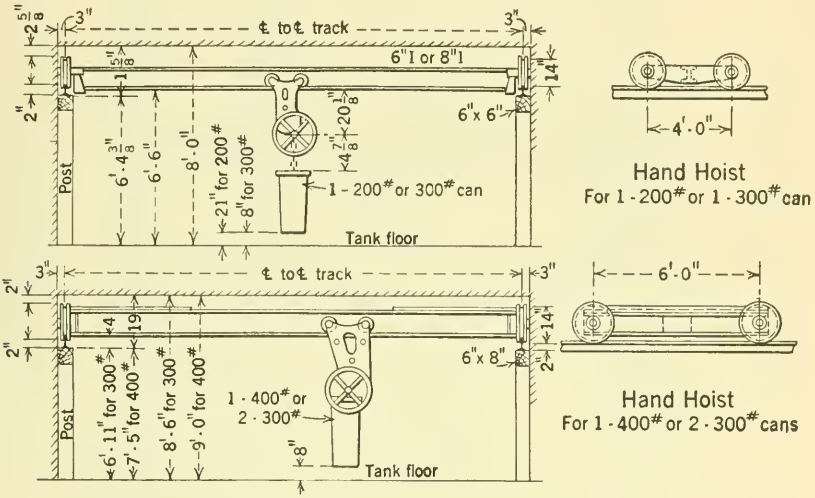



Fig. 271.—Hand-Operated Can Hoist.

where the water for the cans is 70 deg. F. or higher and where suitable care is taken to prevent much loss due to the warming of this water between the fore-cooler and the cans. If the operating conditions were constant the multiple-effect compressor could be used to advantage here, but the conditions are not constant for any length of time. The result is that either a separate machine is used for precooling the water for the cans, or the attempt is made—as in the design above—to increase the suction pressure to as high a pressure as possible.

In choosing the details of the air agitation and harvesting, the costs[16] should be carefully considered, as well as other local conditions. The high-pressure air system was selected for the problem, and also multiple

[15] See Chapter II.

[16] See first part of the chapter.

can lifting and filling devices. These matters should be given careful consideration. Figure 271 gives details for the hand-operated hoist. At

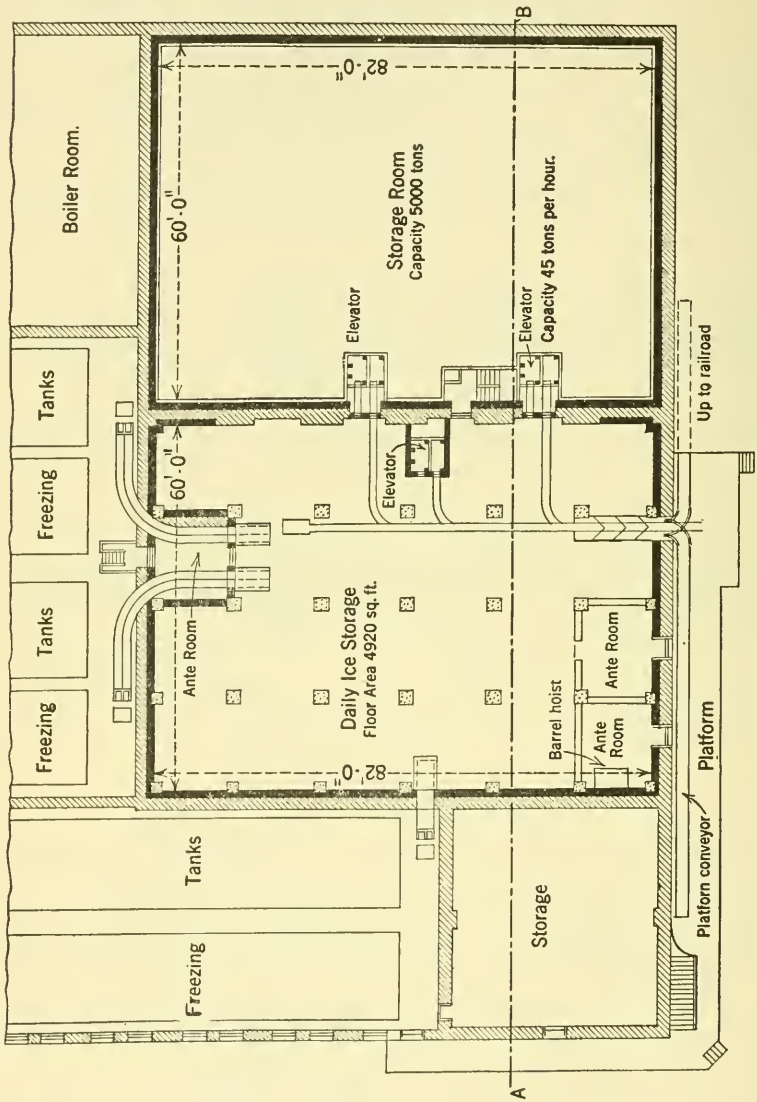



Fig. 272a.—Typical Modern Ice Plants.

the present time this type would be used only for the smaller plants. The larger plants would use the *air* or the electric motor-driven hoist.

**Ice-making Plants.**—Figures 272, 273 and 274 give details of modern ice-making and ice-storage plants. All plants have a *daily* ice storage,

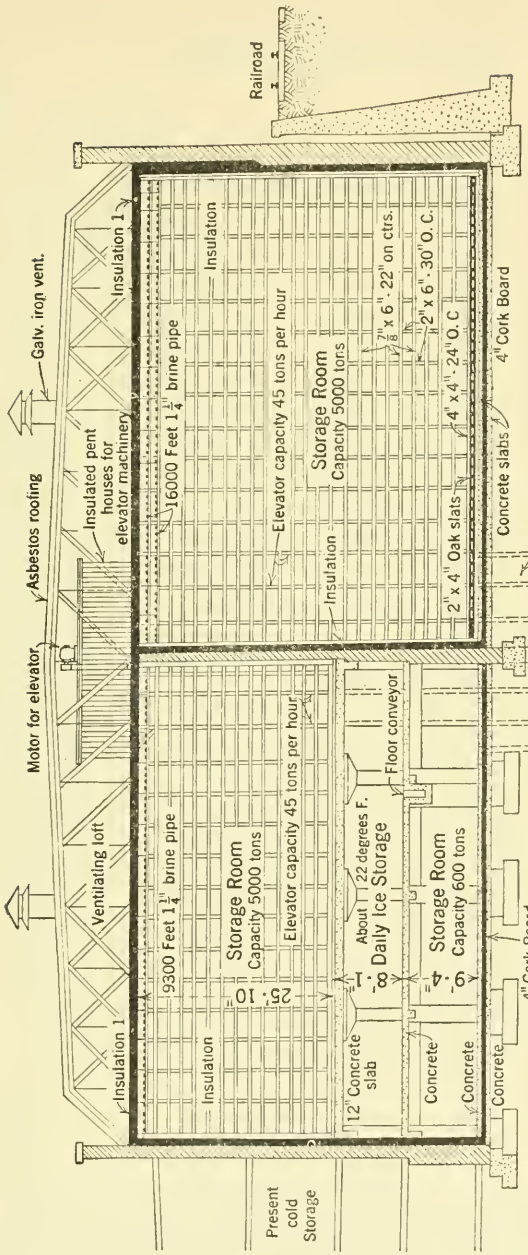



Fig. 272*b*.—Typical Modern Ice Plants.

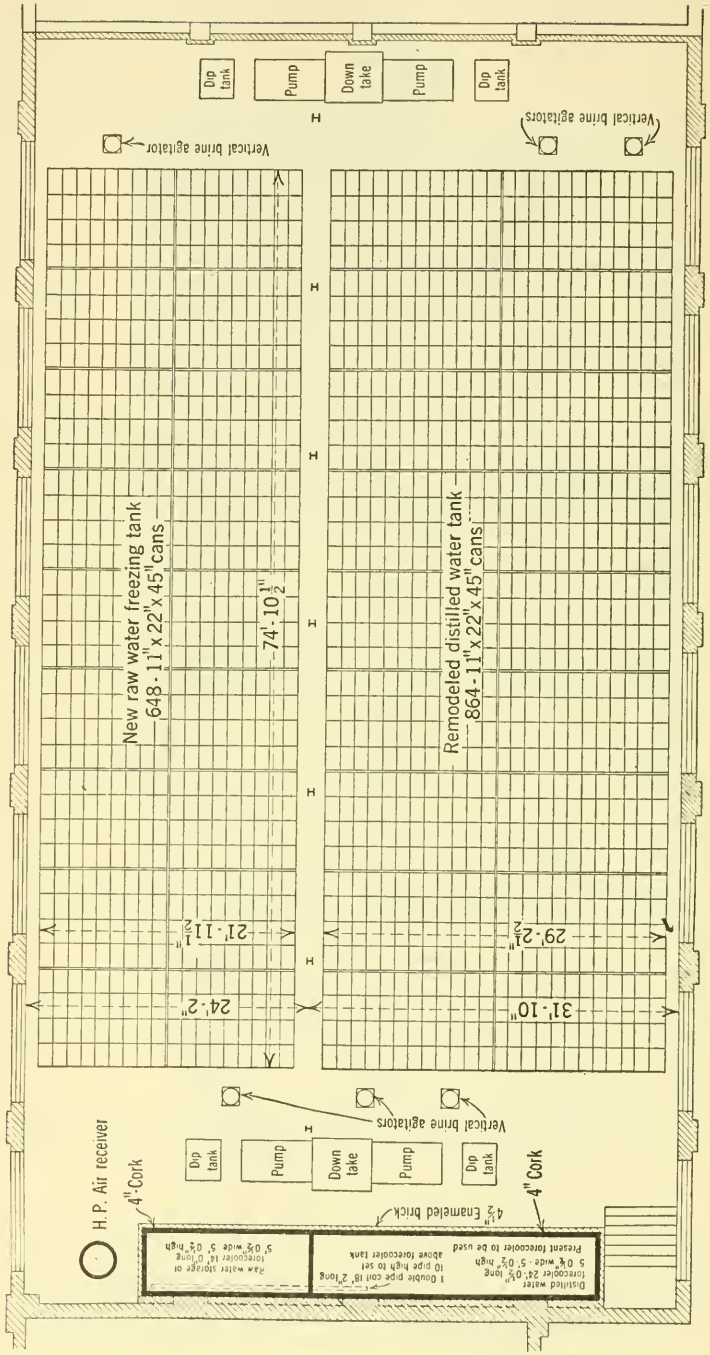

Vertical brine agitator
Vertical brine agitators
Dip tank
Pump
Down take
Pump
Dip tank
New raw water freezing tank
—648 - 11" x 22" x 45" cans
74' - 10½"
Remodeled distilled water tank
—864 - 11" x 22" x 45" cans
21' - 11½"
29' - 2½"
24' - 2"
31' - 10"
Vertical brine agitators
Dip tank
Pump
Down take
Pump
Dip tank
4" Cork
4" Cork
4½" Enameled brick
H.P. Air receiver
Distilled water tor-coo-ler 24' - 0½" long 5' - 0½" wide - 5' - 0½" high
Present tor-coo-ler to be used
1 double pipe coil 18' - 2" long 10 pipe high to set above tor-coo-ler tank
Raw water storage or tor-coo-ler 14' - 0" long 5' - 0½" wide 5' - 0½" high

Fig. 273*a*.—Typical Modern Ice Plants.

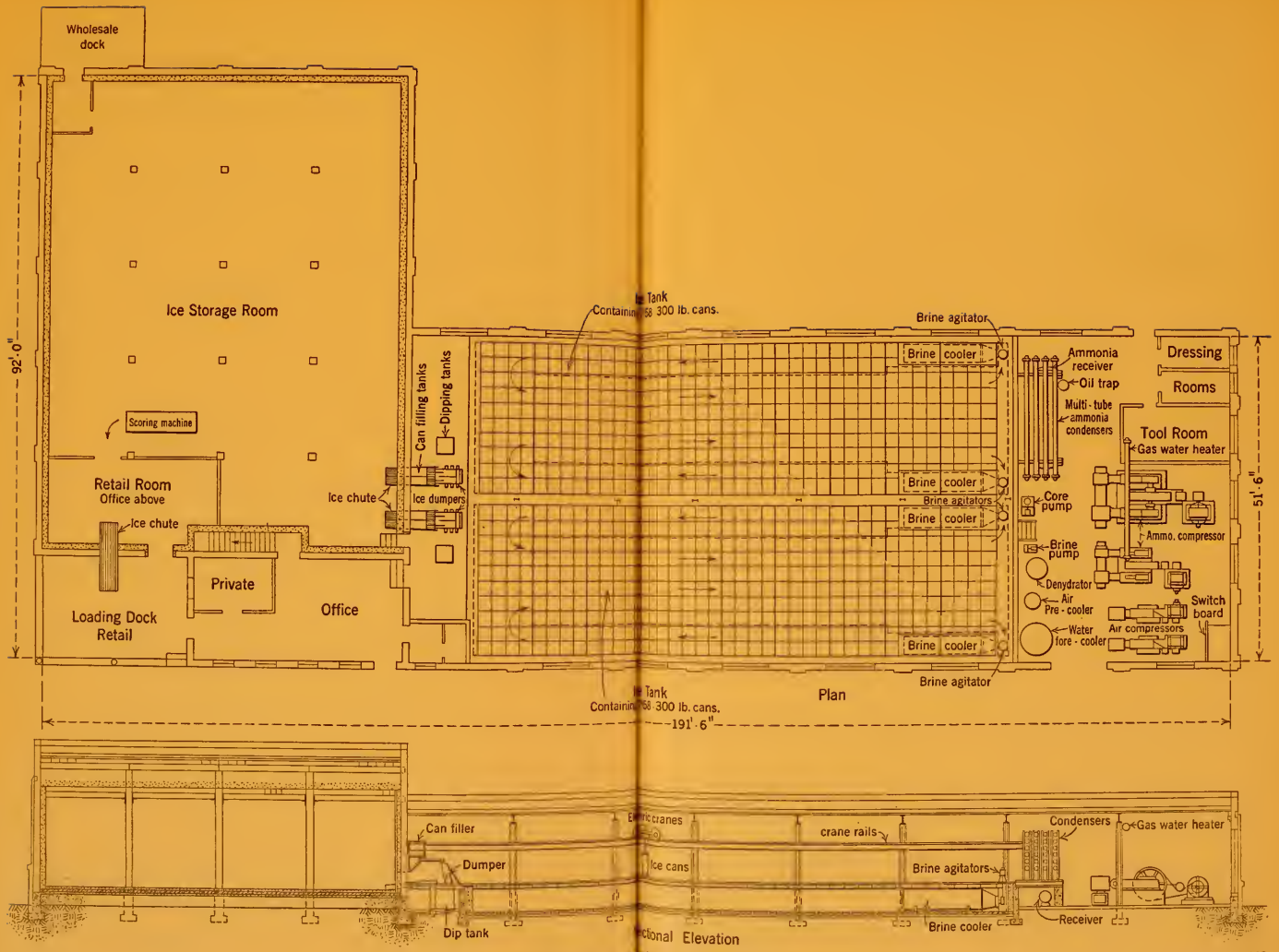



FIG. 274.—Typical Modern Ice Plants.

*To face page 423.*

and most of them have a storage sufficient in capacity to permit an emergency shut-down of from three to four days. It is a question whether the large ice storage warehouse, using a smaller sized compressor for operation from ten to eleven months of the year is more economical than the larger sized compressor and ice tank, to be operated as the demand for ice requires, without the ice storage.

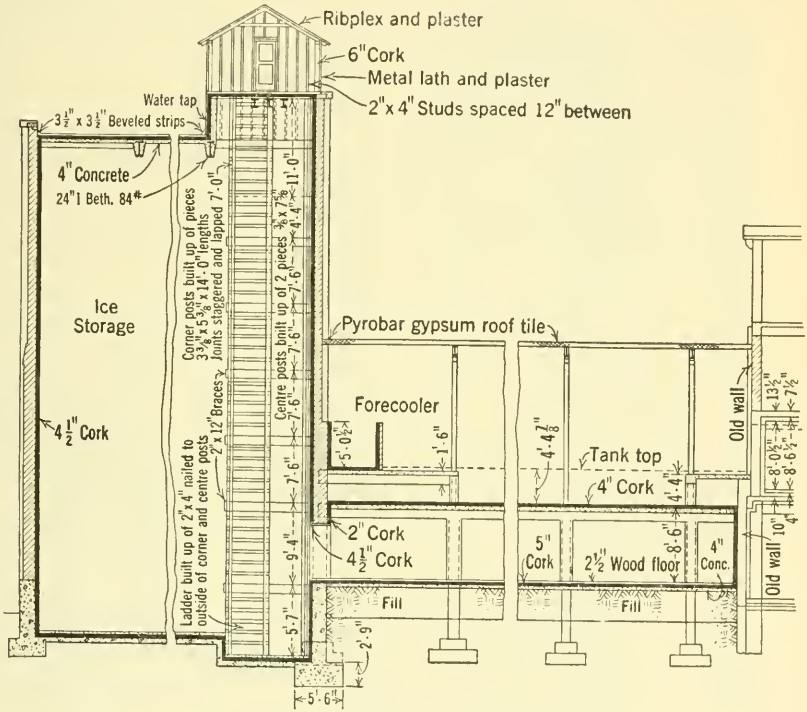



Fig. 273*b*.—Typical Modern Ice Plants.

The following paper, by H. T. Whyte, assistant general manager, engineering and refrigeration, of the Pacific Fruit Express Co., San Francisco, Calif., was presented at the Fifteenth Annual Convention and Exhibition of the N.A.P.R.E. on November 12, 1924.

The capacity of compressors, tanks and storage rooms of plant should be in direct relation to the annual tonnage requirements and, as well, the daily and monthly demand. For instance, where a large daily demand is encountered, which goes on for a period of five months or more, it has been found as a rule more economical to provide for practically the entire requirements through machine capacity, with but small storage room for reserve, in that greater economy can be obtained through pulling requirements direct from tanks, rather than providing large storage room and storing ice throughout the year for disposal during summer months, resulting in excess cost on account of extra handling and, as well, the cost of maintaining

temperatures on storage room throughout the year, and with practically no shut-down period during the winter months.

For example, let us compare the relative merits and costs of operation of a 200-ton daily capacity manufacturing plant, costing approximately $300,000, with a 125-ton daily capacity manufacturing plant, with 10,000 tons storage room, costing $277,000 where the annual demand would be 25,000 tons.

Aside from the saving indicated above (loss account of breakage in storage and

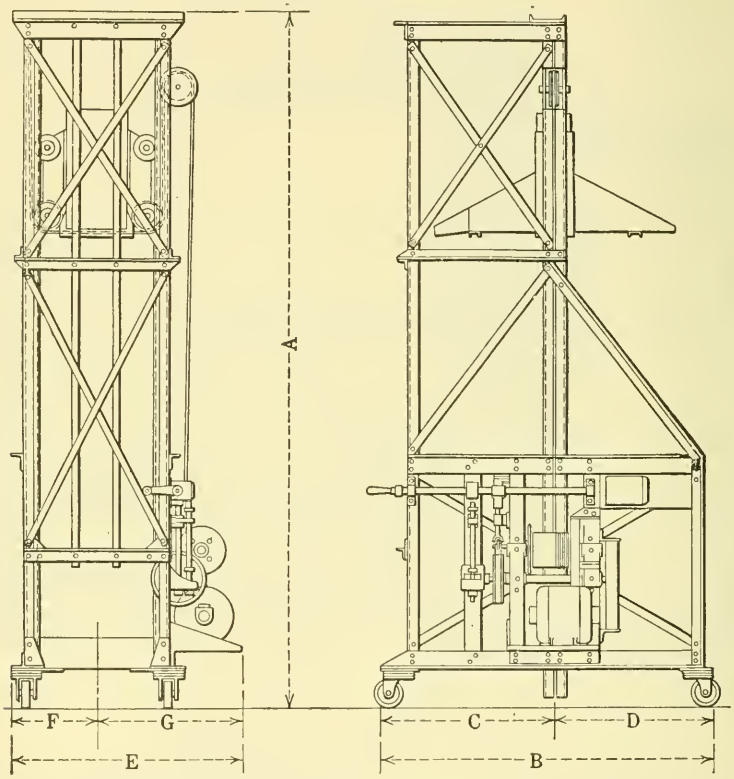



Fig. 275.—The Benching Machine.

removing from storage would be practically eliminated), there is the advantage of supplying your patrons with first quality fresh ice, with nicely squared edges as against stored ice. However, there is the disadvantage of not being able to run continuously even during the peak load period, on account of weather changes which might result in having to shut down part of the plant for a few days, which of course is quite costly.

Climatic conditions will have considerable bearing on the type of installation, with particular reference to type of condensers, and whether cooling tower would prove desirable; also, in the type of construction to be used for tank and compressor room building. In some locations, it is possible to construct this portion of the plant very cheaply, yet serviceably.

Water is one of the most important factors affecting ice manufacturing production costs, for the reason that pure water freezes more rapidly, makes tougher ice, with consequent less breakage and smaller loss due to slush. Also, cold water requires less forecooling for ice making and less is used for condensers; it reduces head pressure with consequent reduction in power consumption. Ample supply of water should also be assured, for lack of sufficient water over condensers is as bad as warm water, in that proper cooling and consequent reduction of condenser pressure cannot be accomplished unless good and sufficient quantity of cold water is available.

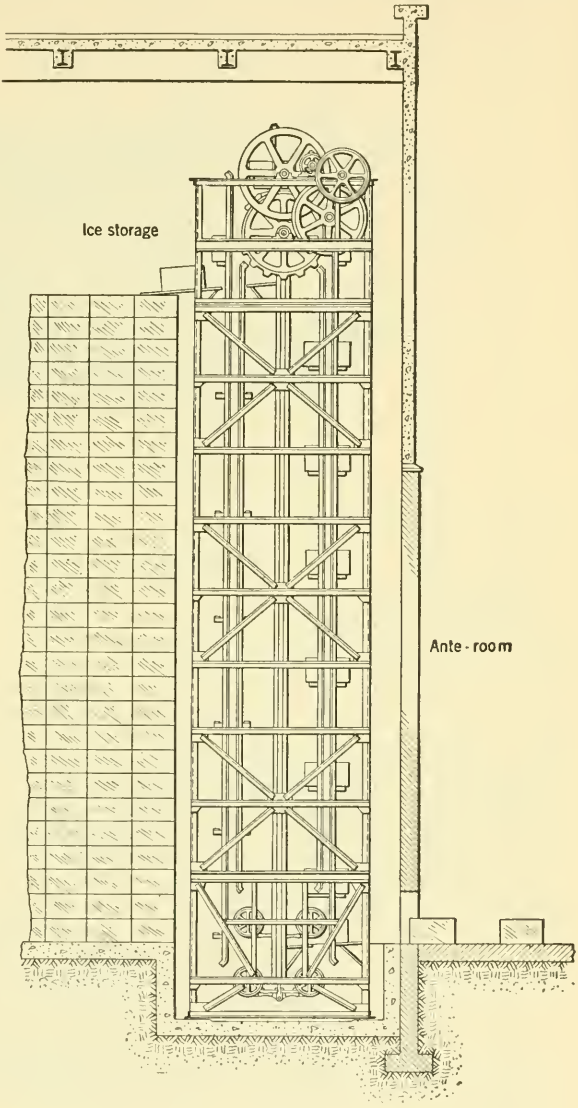



Fig. 276.—Ice Elevating Device.

**Ice Storage.**—In ice storages two designs of elevating and lowering machines may be used. The daily storage usually has the ice on end, and just one tier. A chain conveyor nearly flush with the floor is used in the daily storage for horizontal moving of the ice. For ice storage rooms of medium height the benching machine (Fig. 275) and for taller storage rooms, as in Fig. 273, a device more like Fig. 276 would be used.

**Ice Scoring.**—With certain retail trade there is a demand for scored ice. By this is meant that the 300 and 400-lb. ice cakes have had the

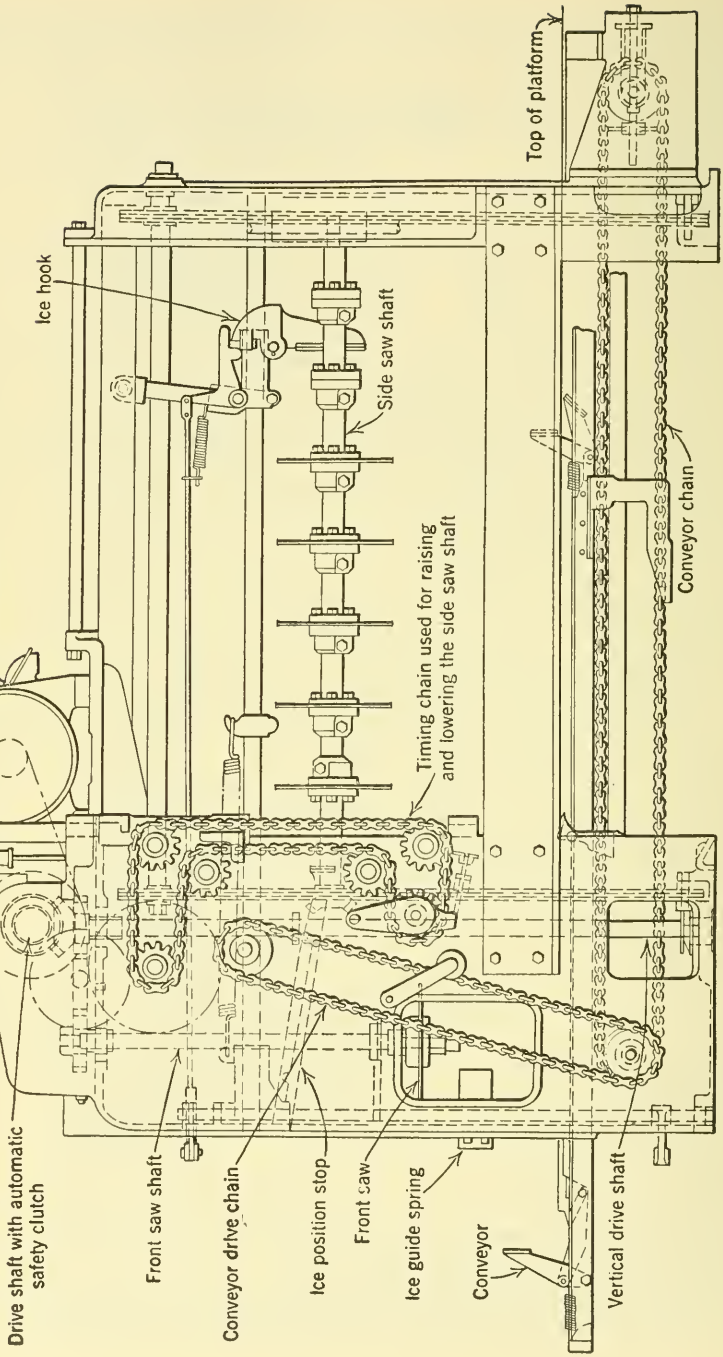



FIG. 277.—Ice Scoring Machine.

proper saw cut so as to register the size of a 25, 50, 75, or 100-lb. piece of ice. Figure 277 shows such a scoring machine.

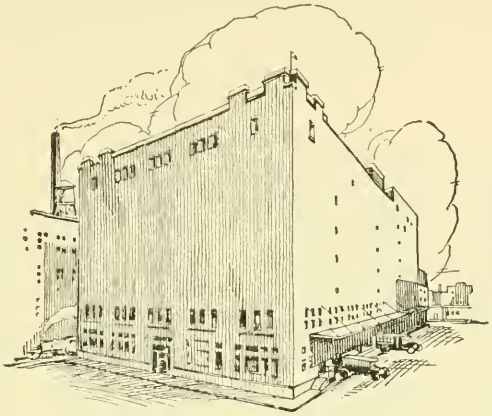


FIG. 278.—Cold Storage Warehouse.

| Item | 10,000 Tons of Storage | | | |
|---|---|---|---|---|
| | 200-ton plant | | 125-ton plant | |
| | Cost per ton | Total | Cost per ton | Total |
| General expense, including insurance, taxes, depreciation and interest on investment (14 per cent) | $1.68 | $42,000 | $1.55 | $38,780 |
| Plant labor | .10 | 2,500 | .14 | 3,500 |
| Tank room labor | .10 | 2,500 | .14 | 3,500 |
| Day storage room labor | .06 | 1,500 | .10 | 2,500 |
| Power | .45 | 11,250 | .47 | 11,750 |
| Ammonia | .05 | 1,250 | .05 | 1,250 |
| Water and other supplies | .05 | 1,250 | .05 | 1,250 |
| Maintenance | .15 | 3,750 | .15 | 3,750 |
| Salary of portion of crew during shut-down period | .15 | 3,850* | .088 | 2,200† |
| Winter storage room labor, 10,000 tons | ..... | ..... | .40 | 4,000 |
| Winter storage room refrigeration, 10,000 tons | ..... | ..... | .30 | 3,000 |
| Totals | | $69,850 | | $75,480 |
| Average cost per ton | $2.79 | | $3.02 | |

Saving in favor of large daily capacity, $0.23 per ton.

* On basis of 5 months' operation; shutdown 7 months @ $550 per month.
† On basis of 8 months' operation; shutdown 4 months @ $550 per month.

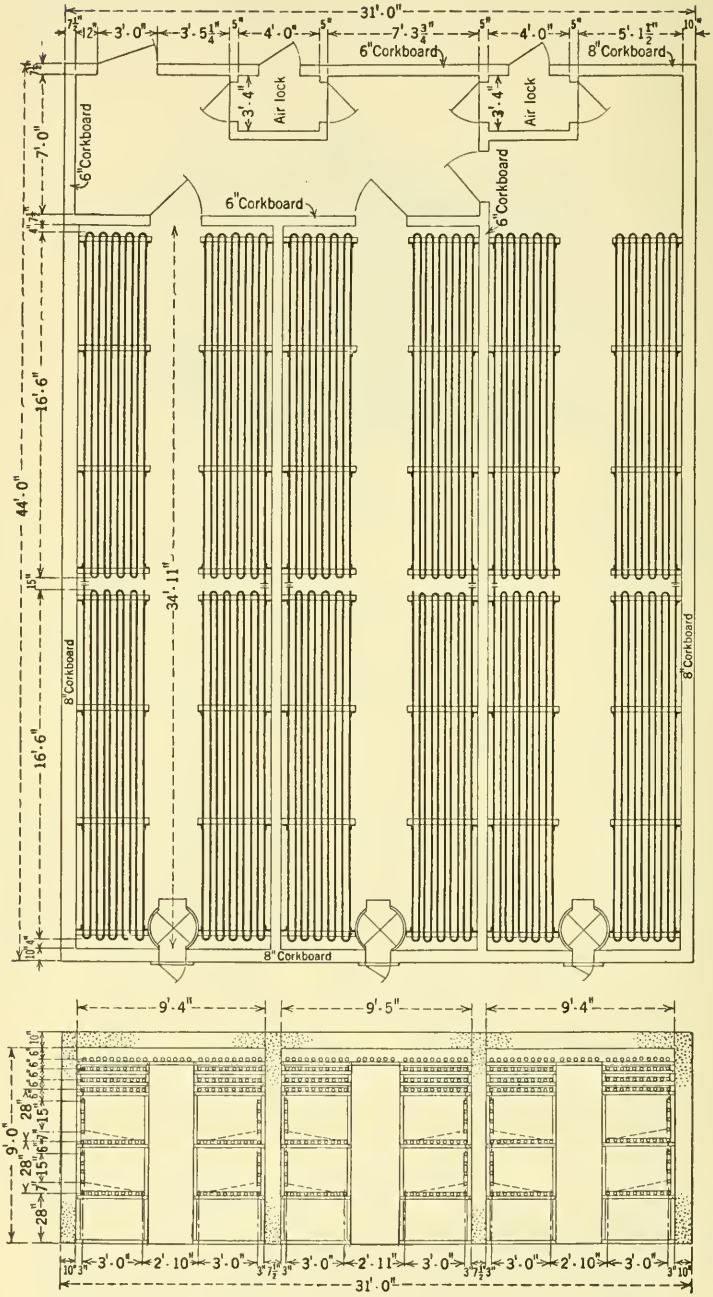



Fig. 279.—The Sharp Freezer.

## CHAPTER XIV

## COLD STORAGE

The most vital application of mechanical refrigeration is cold storage. The space in the United States cooled mechanically to 40 deg. F. or lower for all kinds of cold storage work, exclusive of the packing house, is estimated at about 700,000,000 cu. ft., and this represents about $250,000,000 of invested capital. No estimate of the amount of goods handled in this space is accurately available. There are few industries so vitally concerned in the life of the nation, touching as this does the staple foods such as meat, poultry, fish, vegetables, fruits, eggs and dairy products.

### THE COLD STORAGE INDUSTRY (1925)

By A. W. Oakley

Some thirty-five years ago the cold storage industry carried along in a limited way using natural ice as a refrigerant, but in the past twenty-five years there has been rapid development in the science and art of refrigeration and its adaptation in a commercial way to the preservation of food products from the producer to the consumer. This is indicated by figures giving the approximate refrigerated space in public cold storage warehouses, which in 1914 was 200,000,000 cu. ft. and in 1923, 298,000,000 cu. ft., as given by the United States Department of Agriculture, whereas in 1922 the combined public and private warehouses, including meat-packing establishments, aggregated 559,000,000 cu. ft., of which space about 25 per cent is freezer and 75 per cent cooler. In this space there is carried approximately 1,930,000,000 lb. of apples, cheese and eggs at cooler temperatures; 275,000,000 lb. of butter, poultry, and frozen eggs at freezer temperatures; 75,000,000 lb. of frozen fish and about 1,140,000,000 lb. of meat and packing-house products carried in the cold storage warehouses of meat-packing establishments making a grand total of 3,420,000,000 lb. of products mentioned. These figures do not include such products as dried fruits, nuts and miscellaneous fruits and vegetables and other articles which are not reported by the United States Department of Agriculture, and for which no statistics are available but which represent a very considerable amount. This, with the constantly increasing demand of the small individual refrigerator, as used by the retailer, made possible by automatic machinery, serve the marketing and distribution of perishable foods.

Cold storage warehouses serve to stimulate production and prevent gluts in the market, and the modern warehouse is constantly striving for better handling and preserving of perishable goods in the cold storage rooms by proper use of ozone, ventilation, air circulation and control of humidity. Cold storage warehouses are a necessity as a reservoir in feeding the population in big cities, as some 90 per cent

of the food comes from other states and distant countries. Some idea may be obtained by the following carloads of perishables consumed in New York City per week:

| | |
|---|---|
| Dairy Products | 800 |
| Milk | 2200 |
| Meat | 750 |
| Live Stock | 600 |
| Poultry | 300 |
| Fruit | 850 |
| Vegetables | 1750 |
| Sea Food | 50 |
| Total | 7300 |

These commodities are handled by some 60,000 distributors and 400,000 employees. In New York City, with its congested business section, one office building alone housing 10,000 business people, most of whom consume one meal near business, means the providing of 30,000,000 meals per day by New York markets.

The cold storage warehouse also provides against food shortage in emergencies such as railroad strikes and embargoes.

The object of cold storage is an economic one. It provides a means of extending the season and continues the market for vegetables like celery, lettuce, onions, potatoes, cabbages, etc.; for fruits, like apples, pears, plums, oranges, lemons, etc.; for butter, cheese and eggs, and for raw and cooked meats. The purpose of the industry is to retard the natural processes of decay, this delay being accomplished best by freezing, protecting the surface (such as with fish) by a glaze of ice, or by holding the article as close to the freezing point as the nature of the commodity will allow: as for example, apples, which may be kept at 28 or 29 deg. F. Eggs are a glut on the market in the spring, and are scarce and expensive in the fall and the winter. Fruits like plums, strawberries, loganberries, etc., must be canned in order to extend the season over the natural one of a few weeks, unless cold storage can solve the problem.

To serve a useful purpose, then, the cold storage warehouse must take the surplus supply—which must be of the best quality always—and hold it until the demand increases. This means that only under exceptional conditions would the duration of the storage exceed 9 to 10 months, and according to the United States Government reports the average length of storage for various articles is:

| | Months |
|---|---|
| Eggs | 5.91 |
| Dressed poultry | 2.42 |
| Butter | 4.43 |
| Beef | 2.28 |
| Mutton | 4.45 |
| Pork | 0.88 |

In addition, it is estimated that apples are kept about 5 months, fish (frozen) and dried fruits approximately 6 months.

To give an idea of the relative quantities, it is estimated that the average large cold storage warehouses in the larger cities devote one-quarter of the space to egg storage during the egg season, about 10 to 12 per cent to meats and fish, one-quarter to one-third to fruit storage, and about one-fifth to one-third to butter and cheese. The remainder of the space in the *public* cold storage warehouses is used by a large miscellaneous variety. At special periods, of course, these proportions are very different, as, for example, apples may take half of the space in the fall, and eggs half of the space in the spring.

**The Apple.**—Although apples do not freeze until the temperature is close to 27 deg. F., it is recommended[1] that the temperature be not lower than 30 deg. F., and that 30 to 31 degrees gives the most satisfactory results, both as regards the holding of the fruit and the condition following the removal from cold storage.

The humidity of the air in the cold storage room is very important (for apples and all commodities held in storage) and only under exceptional conditions is the air too moist. As a rule the *wilting* of the fruit due to the drying out of the moisture is one of the most serious problems, but packing in air-tight boxes or barrels or wrapping each apple will decrease this loss. The usual relative humidity of 80 to 90 per cent with an average of 85 per cent gives good results.

According to Dr. Magnus, United States Government research as to the effect of *fresh* air ventilation has shown no advantage from this. In tests there was apparently no difference in the rate of ripening, in the appearance of the fruit, or in the quality of the apples so ventilated as compared with similar lots not so ventilated. There was no appreciable difference in the amount of sugar or amount of acid, and it was impossible to distinguish any difference in the aroma in such fruit on removal from storage.

In order to prevent scald,[2] however, there must be a lively *circulation* of air. This means that air must be able to reach every individual apple and carry away the heat generated by the chemical action which is always going on to a greater or lesser degree. Scald is caused apparently by the accumulation of certain gases given off by the apples themselves, and it can be prevented only by air circulation or by the absorbing of these gases. An oiled (not waxed) wrapper with 15 per cent by weight of oil decreases scald to a marked degree. Apples scald far less in crates

[1] Dr. J. R. Magnus, Handling of Apples and Pears in Storage, U. S. Bureau of Agriculture.

[2] Farmers Bulletin, No. 1160, U. S. Department of Agriculture.

and boxes, while barrels with 15 holes $\frac{3}{4}$ in. by 4 in. cut in the staves decreased the scald by permitting the air to circulate freely provided the boxes or barrels are properly stored. If the apples are in boxes they should be at least $\frac{1}{4}$ in. apart, and have 1-in. strips between layers. If the apples are in barrels they should be down and not on the ends. The best practice is to stow 6 to 8 high using 2-in. by 2-in. strips at the heads to stimulate natural circulation. Apples are not improved by moving after stowing until ready to ship. The amount of carbon dioxide permissible is not definitely known, but according to experiments in Cambridge, England, 12 per cent of carbon dioxide was found to be an aid in the keeping of the apples, but more than 12 per cent decreased the holding power.[3] British practice advocates the use of the *ozonator* although this is not generally employed in the United States.

**The Pear.**—According to Dr. J. R. Magnus, the best storage temperature of the pear is 30 deg. F. Handling, storing, air circulation and humidity are practically the same as for the apple. Rapid cooling of the fruit does not injure it, in fact the more rapid the better. Blackening of Bartlett pears is shown by experiment to be due to their being picked too early, in consequence of which there is not the natural skin wax which is a protection. Blackening of pears and scald in apples appear to be similar conditions. In Bulletin No. 377 of the University of California, by Overholser and Latimer, the results from investigations concerning the effect of temperature, the degree of maturity, region where grown, etc., on scald and blue mold are given.

**The Egg.**—According to Bulletin No. 729, United States Department of Agriculture, only large eggs that are clear and have clean whole shells should be used for long holding in cold storage. The net weight of a case (30 dozen) of eggs packed for storage should be not less than 42 lb. They should be packed in new, odorless cases fastened preferably with cement-coated nails with medium heavy fillers. Odorless fillers should be used on the top and bottom, and the lid should be securely fastened. (See Table 97 for operating conditions.)

The egg room should be dry, clean and free from odors, and the temperature should be kept at from 29 to 32 deg. F. In storing, ventilation must be provided, using 2-in. strips on the floor and $\frac{1}{2}$-in. or thicker strips between the cases in the stacks. If the eggs are fresh and have clean whole shells they may be successfully preserved for from nine to ten months. The loss in spring-packed eggs by candling is about 3 to 4 per cent., but it is much greater for summer-packed eggs. The loss in weight due to shrinkage per case is about 3 to 4 oz. per month. The relative humidity of the room should be about 85 per cent.

[3] A. H. Ashbolt (Agent General for Tasmania), Cold Storage, 1923.

In the summary of Bulletin No. 775 (United States Department of Agriculture) it is stated that clean whole eggs that have not been wet show a negligible loss after 10 to 11 months in storage. Imperfections in handling, grading and marketing previous to storage are responsible for most of the bad eggs. Commercial selection by inspection is stated to be very inefficient. Average tests showed $17\frac{1}{2}$ cracked and one leaking egg per case. Dirty and spoiled eggs were included. Candling is by far the preferred method, and by its means the egg can be graded according to quality, and cracked eggs can be detected. There usually are about 3 eggs cracked per case of spring firsts. If eggs are dirty, cracked, leaking, heated or stale they should be packed in bulk and frozen, under which condition they will keep a year or more. Eggs fresh when stored show after storage an increased air space and a yellowish white. The percentage of ammoniacal nitrogen is an indication of the decomposition. The cold storage taste present after the seventh month seems to be an absorption of the odors from the strawboard fillers or other surroundings.

Dr. Mary E. Pennington,[4] speaking before the American Association of Ice and Refrigeration (1923), said, in explaining certain research in egg storage that aeration with fresh air had a deterrent action on the absorption by the eggs of storage flavors when the present egg package is used and when the temperature carried is from 29 to 31 deg. F. In this research the air was cooled and conditioned by the use of water and brine sprays, and mold appeared to be checked by the use of rapidly moving fresh air, whereas the fact that the air was in a lively circulation did not increase the shrinkage, provided the proper humidity was maintained.

The freezing of eggs in bulk is good economy. It withdraws from the

[4] Dr. Mary E. Pennington, American Association of Ice and Refrigeration, 1923. The conclusion of this paper is as follows:

While the foregoing report is based on but one experiment, and one in which June instead of April or May eggs were under observation, findings indicate that:

1. Aeration with fresh air had a deterrent action on the absorption by the egg of storage flavors when the present standard egg package is used, and when the temperature is from 29 deg. to 31 deg. F.

2. The humidity of the storage rooms was controlled during the late summer and winter months by a regulated current of outdoor air.

3. Apparently moving air does not accelerate the usual shrinkage of stored eggs, provided the air be sufficiently humidified.

4. It is of interest to note that a well started growth of "whiskers," and a beginning growth of mold, was checked by a rapidly moving current of fresh air.

5. The total loss in weight of the eggs averaged approximately 0.5 per cent per month. This is very similar to the loss observed by investigators in the U. S. Department of Agriculture studying the commercial preservation of eggs by cold storage.

market surplus quantities of edible eggs of a grade below first, and makes them available for use the year round. They are put on the market in the form of whites, yolks, or a mixture of whites and yolks. The 30-lb. tin is the usual package. Frozen eggs should be frozen promptly, and should then be kept at a temperature of 10 deg. F. or lower.

**Butter.**—Butter should be carried at 2 deg. F. or lower to secure the best results. It is usually packed in the 63-lb. tubs, the 63 to 78-lb. cubes and the standard boxes for the one pound prints. Butter should be made of cream of limited acidity if it is to be stored for several months, should be pasteurized, cooled promptly and churned without further ripening. Low percentages of salt in the butter are better for keeping than high percentages. Stowing is important as regards air circulation and inspection. The cube and box packages should be separated by at least one inch dunnage. D. C. Dyer says,[5] that oxidation of the fat itself is not the cause of undesirable flavors in cold storage butter held at zero deg. F., but is due to a chemical change in one or more of the non-fatty substances found in the buttermilk.

**Poultry.**—Dressed poultry, properly prepared for storage, fresh, absolutely free of any visible or olfactory signs of decomposition and correctly handled may be maintained in good quality for 12 months. It is not advised that longer periods be used, although it is possible, except that the palatability is decreased.

For the best results the poultry should be dry picked and dry packed. Poultry which has been in contact with ice should never be stored. Ice-packed poultry and scalded poultry usually age more quickly. Poultry should be packed in clean, well-made boxes—usually 12 birds to the box—in either single or double layers. Each fowl should be wrapped separately and the box lined with a suitable paper. The box should be tight and the packed boxes should be placed in the sharp freezer and stored so as to permit air circulation. During freezing the room temperature should be from 0 to 5 deg. F., and afterwards it should be held at 15 degrees or lower.

**Fish Storage and Freezing.**—Fish is one commodity which must be frozen in a special manner in order to be kept in storage for any length of time without serious change of food value or flavor, yet without freezing fish would be practically unknown in inland communities, and such fish as bluefish would be in the market for a few weeks only, while salmon would be unknown, except on the sea coast, in any way but as cured fish. The market for mackerel, halibut, pike, smelts, etc., would be curtailed considerably, and fish of many varieties would be a costly delicacy.

[5] Journal of Agricultural Research, September, 1916.

For storage fish needs ice "glazing." By this is meant that a protective coating of ice be formed on every individual fish or around a number of fish if the fish are small. Unless they are glazed the skin is likely to turn white[6] and the fish to shrivel, because of the loss of moisture. The glaze also tends to prevent the shrivelling of the noses and the fins, to keep the eyes from becoming opaque and the gills (normally red) from becoming dark and brown, and in general, provides a surface on which molds and fungi cannot grow.

As a rule the following procedure is considered good practice. The fish are first washed free of dirt and slime and the larger fish—bluefish particularly as they are heavy feeders—are gutted before freezing. On the contrary, some localities prefer salmon to be frozen in the round. The smaller fish, as the small weak fish, mackerel and salmon, cannot be gutted and are therefore frozen in their natural state in pans containing approximately 40 lb. Large halibut and salmon are frozen separately. The pans are placed directly on the shelving made up of piping (Fig. 279) and the piping ratio is sufficient to permit a temperature of the sharp freezer of from − 5 to − 15 deg. F. The shelving is designed to permit passageway for convenient handling. The fish are reduced in temperature in from 12 to 30 hours, depending on the room temperature and the size of the fish, and as soon as this is done they are ice glazed. The glazing operation is done by having a trough of water at about 32 deg. F. (which is frequently changed) in a room at a temperature of from 20 to 25 deg. F., and the fish are made to pass through the trough. The process is repeated from three to five times or until the required thickness of ice is frozen upon the fish. Another method is to use a grilled platform which is lowered and then raised with the fish. Finally the fish are placed in the ordinary type of cold storage room until used, the room being kept constant at some temperature from 0 to 10 deg. F. After 3 to 5 months the ice glaze will be lost to the air in the room by evaporation, and will need to be replaced. Boxing of the glazed fish is becoming more general, as it simplifies the handling and the shipping of the fish, and permits a more careful record to be kept of the duration of storage. The boxes are lined with heavy Manilla paper, and when so done the fish are protected from injury and the glaze takes longer to evaporate. As a rule 4 to 5 cakes of panned fish, 120 to 150 lb., commonly are packed in one box, the size of which is made just right to take the cakes from the pan. Large fish are usually wrapped first in a vegetable parchment paper, and then are carefully packed in boxes lined with Manilla paper.

Tests by the Bureau of Chemistry, United States Department of

[6] Bulletin No. 635, U. S. Department of Agriculture, by Clark and Almy.

Agriculture (Bulletin No. 635) indicate that properly glazed fish were kept for 27 months during which time and at the end of this period elaborate analysis did not indicate any changes which rendered the fish unsuitable for food, nor were there any important differences in the chemical composition between the stored and the fresh fish. Other experiments show that after 9 months there was nothing in the taste by which the average consumer of fish could detect the stored fish from the fresh fish. However, the average period of storage is about 8 months, as the stock is usually depleted before the fresh fish is again plentiful in the market.

**The Brine Method.**—According to a paper by Dr. Walter Stiles of University College, Reading,[7] the use of cold brine for the freezing of fish was found to be better than freezing by air cooling. All air cooling, according to this authority, caused a loss of surface sheen and brightness due to the loss of surface mucilage, as well as a loss of weight by evaporation of moisture up to 6 per cent. These changes in the fish were not present in the case of brine freezing. The report goes on to show that an ice glaze of 10 per cent increase of weight should last 30 days, but that brine freezing tends to prevent the formation of this ice glaze. J. M. Tabor[8] says that fish plunged into cold brine will freeze in half an hour, that the brine is found to be an antiseptic, and that the freshness of the fish is not affected because the skin is frozen first. Mr. Tabor advises that the fish be cleaned and gutted before freezing. In packing fish "salunol ice" (ice made from water with a trace of sodium hypochloride) was advocated because of its germicide properties.

The freezing of fish in the United States with brine (a patented process) has not become general. The salt does penetrate somewhat into the body of the fish and at times fish so treated is claimed to be unpalatable. The eyes become dulled, and if blood is exposed to the brine it is changed to a dark color. The fish usually have distorted shapes, and pack poorly. But the process is a commercial one and certainly has merit from the theoretical point of view.

**The Peterson Process.**—Mr. J. W. Peterson has developed a process[9] which greatly resembles the manufacture of ice. The smaller fish are placed in scoops or pans of about 28 in. by 18 in. by $3\frac{3}{4}$ to 2 in. deep and then are carefully transferred to No. 15 gage galvanized cans (Fig. 280). These cans are submerged in calcium chloride brine at from $-20$ to $-25$ deg. F., and complete freezing is possible in 2 to $2\frac{1}{2}$ hours, or less. The frame of cans is then lifted out of the brine and is lowered

[7] Report of the Food Investigation Board, 1921, England.

[8] J. M. Tabor, British Cold Storage and Ice Association, 1923.

[9] J. W. Peterson, Refrigeration Engineering, 1924.

into a thawing tank (as in ice making) and when the " cake " is released it is glazed as before. Large fish are packed into the cans individually. It is claimed that the Peterson process offers a low first cost, a small space requirement, that the freezing time is very short, and also that the power and the labor costs are comparable with other methods of fish freezing.

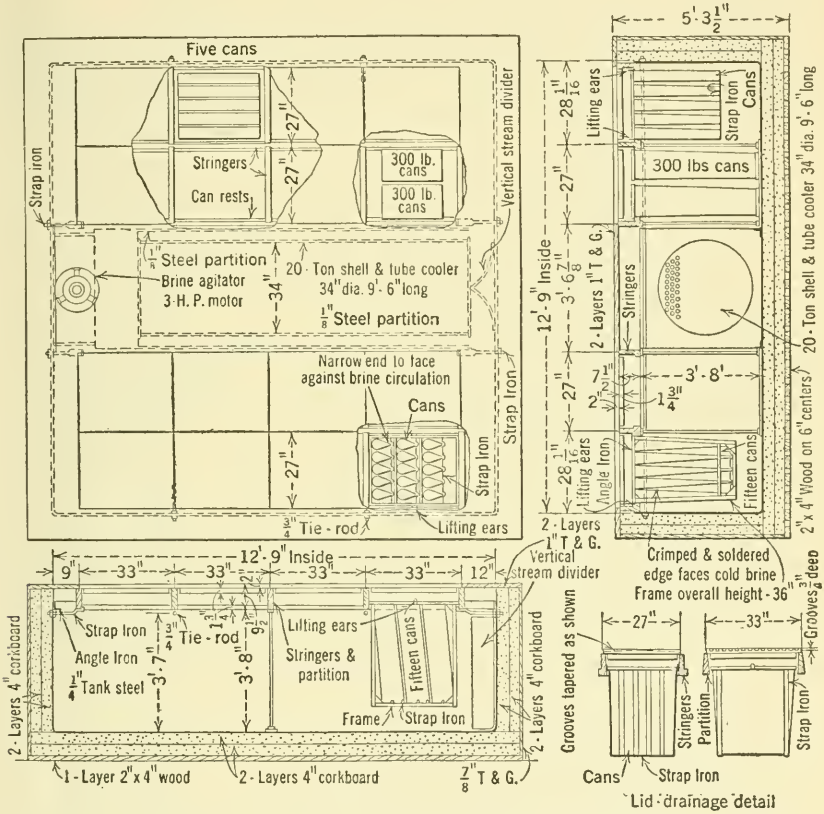



Fig. 280.—Peterson Tank System for Fish Freezing.

**Vegetables:** *Sweet Potatoes.*—Sweet potatoes, free from frost, bruises, and dried sufficiently to remove the surface moisture are stored in bins, crates and hampers, stacked so as to permit good air circulation. They should be " cured " at 85 deg. F. to remove surplus moisture (this requiring from 7 to 21 days) at which time the surface is bright, clear and dry,[10] and then the temperature should be gradually reduced

[10] H. C. Thompson, Storing and Marketing of Sweet Potatoes, No. 970, U. S. Department of Agriculture.

from 50 to 55 deg. F. The room should be kept dry, under which conditions the sweet potato will keep well for from 3 to 6 months.

*Onions.*—Only sound, well ripened onions should be stored—free from loose skins—and properly dried. They should be stored in slatted onion crates, properly stacked for air circulation, or in shallow slatted bins and should be cooled to from 32 to 36 deg. F. as quickly as possible after stowing. A low relative humidity should be carried.

*Cabbages.*—Solid heads, free from injury and loose leaves, should be placed one layer deep on slatted shelves properly arranged for good air circulation. In order to prevent excessive wilting and shrinkage a

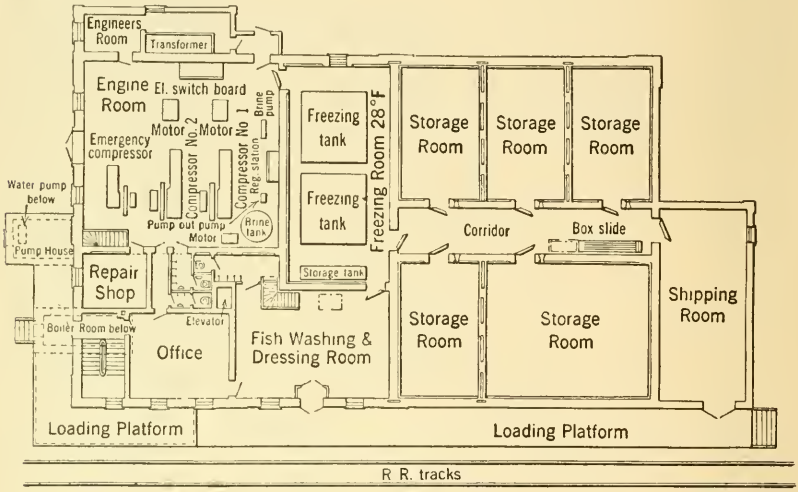



Fig. 281.—The Otteson Fish-Freezing System.

relative humidity of 80 to 85 per cent should be carried, but moisture must not be permitted to form on the leaves.

*Potatoes.*—Potatoes should not be stored deeper than 6 ft., and if in crates or bags good circulation of the air in the room should be arranged for. The air temperature should not be less than 35 nor greater than 40 deg. F. The relative humidity should be as high as possible without precipitation of moisture on the surface, or 80 to 90 per cent. Potatoes should keep 6 months in prime condition.

*Insects in Cereals.*—In a paper by Dr. A. E. Black of the United States Department of Agriculture before the American Association of Ice and Refrigeration, 1922, it was stated that the fruit fly is very easy to destroy, but that the clothes moth larvæ and other pests are not so easily killed. The bean weevil can be killed if reduced to 32 deg. F. for

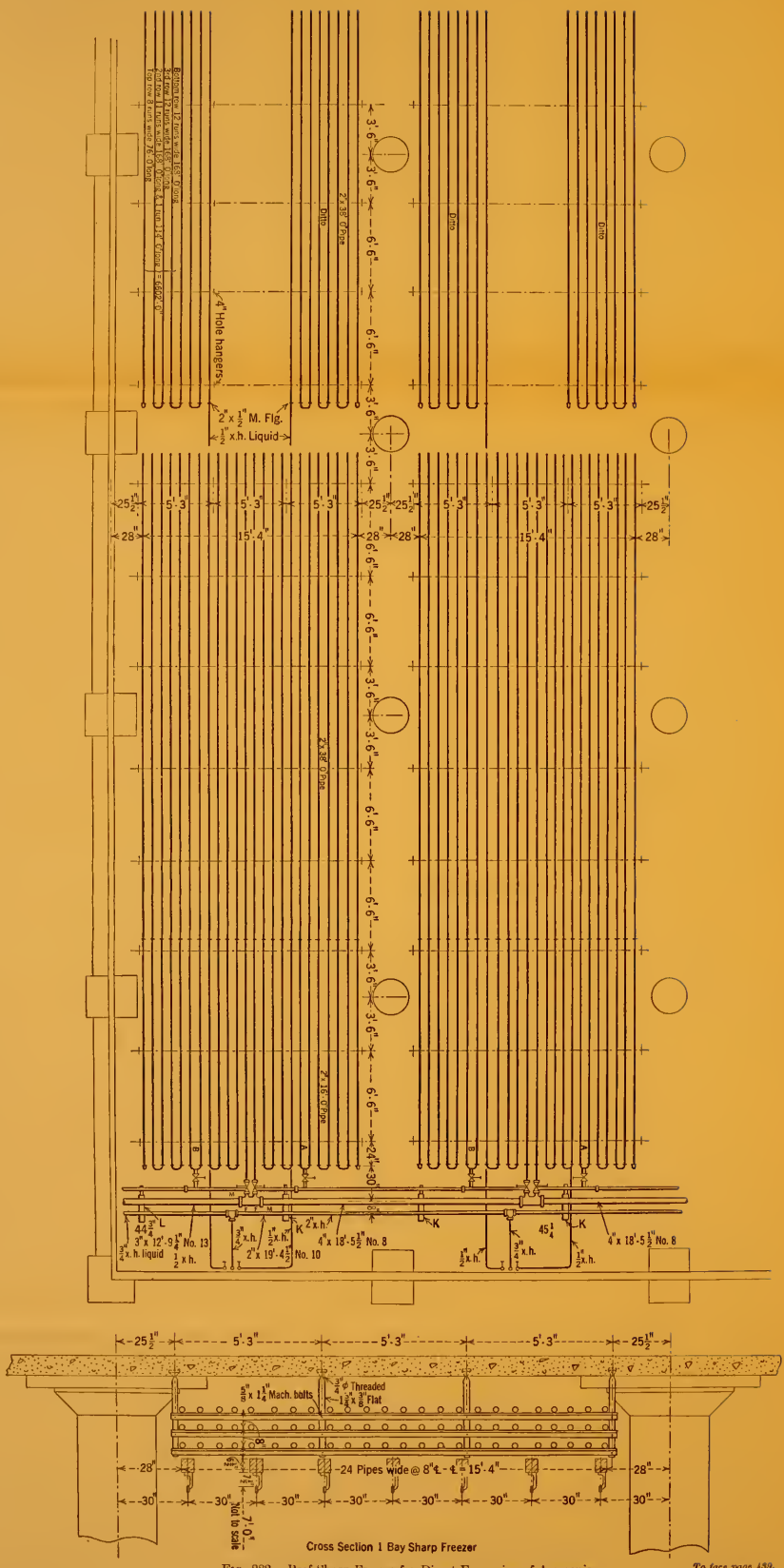



FIG. 282.—Beef Sharp Freezer for Direct Expansion of Ammonia.

*To face page 439.*

56 days, whereas the cow pea weevil will succumb in a month at 32 deg. or even at 39 deg. F. The following insects were subjected to 30 deg. F.

The sawtoothed grain beetle, common in flour and dried fruits, was killed in four weeks. The northern flat grain beetle dies in a week, and the southern variety in two months, whereas the broad-horned flour beetle required only one week. The Australian wheat weevil required from three to four months, as did also the tobacco beetle which likewise attacks cereals. The rice weevil egg was killed in one week, the adults in 17 days and the larvæ in four weeks. The granary weevil (somewhat similar to the rice weevil) required three months at 30 deg. F. The eggs were killed in four weeks and the larvæ in four to five weeks.

**Freezing of Fruits and Berries.**—Research in the preservation of fruits and berries[11] by freezing at 8 to 12 deg. F. showed that, in general, soft fruits and berries can be "kept for from 6 to 10 months at 8 to 12 deg. F., after which they are equal to fresh fruit for most purposes." Cherries in open containers become dark in color, but covering with water and freezing prevented this darkening. The use of sulphurous acid resulted in an inferior flavor. Apricots gave excellent results, especially when stored in water, or crushed with an equal weight of sugar. Red raspberries, currants, loganberries and strawberries, untreated, preserved well at 8 to 12 deg. F., but shrivelled slightly. Best results were obtained by crushing the fruit and adding sugar. The flavor was much better than similar fruit preserved by cooking. Grape juice stored at 10 deg. F. was superior to the pasteurized variety.[12]

*Plums.*—E. L. Overholser[12] says that certain varieties of plums keep well at 36 deg. F. and others at 32 deg. F. The storage time averaged from 10 to 12 weeks and gave a marketable plum of from 2 to 7 days. Sorting and packing are prompt. Precooling to 32 deg. F. is imperative.

## SOME EFFECTS OF FREEZING ON MATURE FRUITS OF THE APPLE

By D. B. Carrick, Cornell University Agriculture Experiment Station.

Rapidly frozen apples exhibit a larger amount of discoloration than do similar fruits given four times as long an interval to reach the same minimum temperature. Apples frozen rapidly to varying degrees, when thawed at 0 deg. C. (32 deg. F.) and 22 deg. F. (72 deg. F.) show an equal amount of browning injury. However, slowly frozen apples when thawed very slowly during several days or weeks at 0 deg. C. (32 deg. F.), uniformly reveal more discoloration than if more quickly thawed at 10 deg. C. (50 deg. F.) or higher.

[11] Cruess, Overholser, and Bjarnason, Bulletin 324, University of California Agriculture Experiment Station.

[12] E. L. Overholser, Bulletin No. 344, Agriculture Experiment Station, University of California.

Freezing points of expressed apple-fruit juice, measured by a Beckmann apparatus, are found to be significantly higher than the depression of the cell sap within the normal, unfrozen apple tissue as determined by thermo-junctions. The depression of the cell sap within the tissue frozen to death was much higher than the Beckmann measurements.

Various external and internal symptoms of frozen apples, such as changes in color, the effect of bruising while ice is still present, and the texture and flavor, are described. Some distinctions are made between frozen and over-mature apples.

Since slightly frozen apples are usually more susceptible to *Penicillium expansum* and other fungous organisms, they do not keep so well in storage as does normal fruit. Severely injured apples cannot be held without loss for any length of time, and should either be disposed of immediately or placed in storage at − 20 deg. C. (− 4 deg. F.). In this low temperature the fruit can be kept almost indefinitely, and it is satisfactory as a culinary product but worthless for fresh consumption.

Preliminary results with the Wagener, Baldwin, Rome, and Ben Davis varieties indicate that − 1 deg. C. (30.2 deg. F.), or somewhat below this point is the optimum cold storage temperature for the apple, as compared with 0 deg. C. (32 deg. F.) or above.

A study of extreme temperature is contained in the United States Department of Agriculture Bulletin No. 1133, entitled, " Freezing Temperatures of Some Fruits, Vegetables, and Cut Flowers," by R. C. Wright and Geo. F. Taylor.

Freezing or freezing injury does not always occur when fruit or vegetable products are exposed to temperatures at or below their actual freezing points. Under certain conditions many of these products can be undercooled; that is, cooled to a point below the true freezing temperature of each and again warmed up without freezing and without apparent injury. Certain products under certain conditions may be actually frozen and then thawed out without apparent injury, while, on the other hand, some products are injured if stored at temperatures well above their actual freezing points. Evidence seems to show that different individuals of the same variety and strain when grown under different conditions will have somewhat different freezing points, and that there are also some variations in the freezing points of products of the same variety and from the same lot.

In view of these facts the freezing points given in this bulletin should be considered only as danger points at or near which, either above or below, there is a possibility of freezing injury if exposed for a sufficient length of time. These are temperatures at which it is unsafe to hold produce for any length of time, as serious danger of frost injury exists.

**Cold Storage for Furs and Fabrics.**—Because of the damage from the ravages of moths and beetles, the fur storage industry has developed to quite an extent during the summer months. Storage at freezing temperatures also tends to improve the lustre of the fur so that the fur has a greater value after the warm weather is over and the furs are needed agan for the fall or the early winter.

*Temperatures Required.*—The moth and the beetle egg requires a temperature of 55 deg. F. for hatching. If hatching occurs the insect

eats the fur to secure the grease and the animal juices existing there. The fur constitutes the chief food of the insect and supplies the necessary material for the cocoon of the moth. If the larvæ are hatched the movement becomes active at 45 deg. F. and above, and the fur will be injured; this activity increasing up to 55 deg. F. where normal activity is reached. Temperatures below 45 deg. F. stunt the activity to a point where little or no damage is done; at 42 deg. it has been demonstrated that although the insect may be still moving, it is very sluggish; and at 40 deg. F. all movement is suspended.

TABLE 93

Summary of Averages

| Fruit and Varieties | Temperatures, Degrees F. | | |
|---|---|---|---|
| | Average | Extremes | |
| | | Minimum | Maximum |
| Apples: | | | |
| Summer varieties | 28.44 | 28.12 | 28.62 |
| Fall and winter | 28.51 | 28.21 | 28.87 |
| Bananas (Jamaica): | | | |
| Green { Peel | 29.84 | 29.76 | 29.92 |
| Green { Pulp | 30.22 | 30.10 | 30.58 |
| Ripe { Peel | 29.36 | 29.15 | 29.53 |
| Ripe { Pulp | 26.00 | 25.45 | 26.50 |
| Blackberries: | | | |
| Black varieties | 29.15 | 28.73 | 29.42 |
| White varieties | 28.40 | 28.12 | 28.63 |
| Logan (*Loganberry*) | 29.51 | 29.32 | 29.75 |
| Cherries | 27.81 | 27.56 | 28.25 |
| Cranberries | 26.70 | 26.28 | 26.93 |
| Currants | 30.21 | 30.18 | 30.25 |
| Gooseberries | 28.91 | 28.70 | 29.18 |
| Grapes (eastern) | 28.16 | 27.85 | 28.37 |
| Grapefruit | 28.36 | 28.00 | 28.50 |
| Lemons | 28.14 | 27.89 | 28.47 |
| Oranges | 28.03 | 27.86 | 28.34 |
| Peaches (hard, ripe) | 29.41 | 29.09 | 29.74 |
| Pears (Bartlett): | | | |
| Hard, ripe | 28.46 | 28.06 | 28.70 |
| Soft, ripe | 27.83 | 27.20 | 28.00 |
| Pears (unknown Japanese variety) | 29.39 | 29.34 | 29.53 |
| Japanese persimmons (Tanenashi) | 28.33 | 28.07 | 28.63 |
| Plums | 28.53 | 28.20 | 28.85 |
| Raspberries: | | | |
| Red varieties | 30.41 | 30.12 | 30.50 |
| Black varieties | 28.76 | 28.24 | 28.79 |
| Strawberries | 29.93 | 29.56 | 30.13 |
| Kind and Variety | | | |
| Beans (snap) | 29.74 | 29.65 | 30.06 |
| Cabbage (Early Jersey Wakefield) | 31.18 | 31.06 | 31.34 |
| Carrots | 29.57 | 29.42 | 29.68 |
| Cauliflower | 30.08 | 29.95 | 30.15 |
| Corn, sweet | 28.95 | 28.65 | 29.22 |
| Eggplant | 30.41 | 30.17 | 30.69 |
| Kohl-rabi | 30.02 | 29.74 | 30.22 |
| Lettuce | 31.20 | 31.03 | 31.38 |
| Onions (dry) | 30.09 | 29.69 | 30.24 |
| Peas (green) | 30.03 | 29.67 | 30.25 |
| Potatoes | 28.92 | 28.80 | 29 02 |
| Potatoes, sweet | 28.44 | 28.10 | 28.72 |
| Tomatoes (ripe) | 30.38 | 30.20 | 30.67 |
| Turnips | 30.23 | 30.16 | 30.48 |

TABLE 94

| Product | Handling Factors | | Warehouse Requirements | | | |
|---|---|---|---|---|---|---|
| | Pre-storage handling and condition when placed in storage | Varieties and grades | A—Construction | B—Temperature | C—Humidity | D—Stowing |
| Apples.... | Apples should be picked when well matured but not over-ripe. In all the operations of picking, packing and hauling they should be so handled as to avoid bruising, skin punctures and other mechanical injuries; and they should be so graded as to be practically free from serious injuries caused by insects, diseases or mechanical means. It is essential that they be handled promptly from the orchard to the storage room and cooled quickly. | Only varieties which have a recognized storage period of three months or more should be considered. | Cold storage houses should be so constructed and equipped as to maintain practically uniform temperature and humidity conditions throughout the storage season.<br>Common storage houses should be sufficiently insulated to prevent freezing, and should be provided with the necessary inlet and outlet vents to permit adequate ventilation and temperature regulation. | Cold storage temperature range should be 31 deg. F. to 32 deg. F. for the storage of apples.<br>Common storage temperature should be maintained at from 31 deg. F. to 36 deg. F. after the initial cooling of the fruit. | Humidity range, 80 to 90 per cent. | Apples should be stowed with sufficient spacing to permit of free air circulation, and to render each lot readily accessible for inspection and withdrawal. |
| Potatoes.. | Potatoes should be well matured, and graded to conform to the specifications of the United States standard grades. Seed stock should be certified by a competent inspector. | All varieties harvested in autumn keep well in storage. | The building should be so constructed and insulated as to prevent fluctuations in temperature. Means for ample ventilation should be provided and all unnecessary light should be excluded. | Temperature range from 35 deg. F. to 40 deg. F. | Humidity range, 80 to 85 per cent. | When stowed in bags, boxes or crates, potatoes should be so piled as to permit free air circulation. Bulk potatoes should not be stored to a greater depth than 6 ft. nor more than 60,000 lb. in a single compartment. They should be carefully handled to avoid unnecessary injuries. |
| Potatoes (sweet).. | Sweet potatoes should be well developed, carefully handled to avoid bruising, and should be practically free from damage caused by disease, insect or mechanical injury. They should not be allowed to become chilled or frosted, and when placed in storage the surface should be dry and practically clean. | All varieties grown on a commercial scale. | The building should be so constructed that all light is excluded, and moderate changes in outside temperature will not quickly affect inside temperatures. Wood construction is preferable, and ample means for ventilation control should be provided. | While the potatoes are being stored, and for a period of ten days to two weeks thereafter, or until the potatoes are cured, a temperature of from 80 deg. F. to 90 deg. F. should be maintained. Thereafter, a uniform temperature of as nearly 55 deg. F. as is practicable should be maintained. Ventilation and artificial heat are necessary to control temperature and moisture. | The percentage of humidity should not be so high that moisture is deposited on the walls of the storage house. | When stored in bins, the potatoes should be carefully poured from basket or crate into the bin. To allow free circulation of air, the bins should have slatted sides and floor, and at least 4 in. of air space on all sides. The bin floor should be raised 2 in. or more above the house floor. When stowed in crates, baskets or hampers, the containers should be stacked so as to allow circulation of air, and to avoid the crushing or breaking of the packages and the bruising of their contents. |
| Onions.... | The onions should be well ripened, dry and thoroughly air-cured when stored. Onions intended for storage should be practically free from damage caused by disease, insects or mechanical injury, and from other stock commercially known as culls. | All common varieties of onions, except those of the Bermuda type. | The building should be so constructed and insulated as to prevent fluctuations in temperature, and means for ample ventilation should be provided. | In cold storage the temperature range should be from 32 deg. F. to 36 deg. F.<br>In common storage the same range of temperature should be maintained as nearly as possible, but with the proper ventilation, onions will keep well in common storage at higher temperatures. | Low humidity is desirable. | Onions should be stowed in suitable receptacles, as indicated under "Containers," and should be stacked in such a way as to permit of free air circulation throughout the lot. |

| | | | | | | |
|---|---|---|---|---|---|---|
| Cabbage.. | Cabbage must be of solid heads, practically free from injuries caused by insects and diseases. Heads should be cut with but few, if any, loose leaves adhering, and carefully handled from field to storage house. Special care should be used to avoid bruising and other mechanical injuries. | Danish Ball Head, or sorts with similar form and texture. | Well ventilated, frost proof root cellar or warehouse type of construction, with ample intake and outlet vents for quick cooling and ventilation, and equipped with slatted shelves supported on staging, so that the heads may be stored one layer deep, with at least 15 to 18 in. clear space around the walls of the building. The ceiling should be so constructed as to prevent drip on the product. | Temperature range, 32 deg. F. to 36 deg. F. | The humidity should be maintained as high as possible without actual deposition of moisture on the product. | Cabbage should be stored on slat shelves in single layers. The height of the staging and the number of shelves will be determined by convenience and dimensions of the building. |
| Eggs..... | Eggs should be moved quickly from the producer to the warehouse. They should be carefully sorted and candled, so that none showing mechanical defects or noticeable deterioration is included in the storage stocks. No washed eggs should be stored. | The grades should conform to those generally adopted by the wholesale trade, until United States standards are promulgated. | Cold storage houses should be so constructed and equipped as to maintain practically uniform temperature and humidity conditions required for successful storage throughout the storage season. | Temperature range, 29 deg. F. to 32 deg. F. | Humidity range, 82 to 85 per cent. | Egg cases should be stowed so that separate lots may be easily inspected, and with ½ in. to 1 in. dunnage between the cases to insure space for free air circulation. |
| Frozen eggs | Eggs should be removed from shell in chilled, sanitary surroundings, and frozen immediately on fish-shelf sharp freezers. | One grade for food. One grade for manufacturing purposes. | Same as for eggs. | Temperature range, 0 deg. F. or below to 10 deg. F. above. | The usual humidity at the temperature of storage. | Protect eggs from heat leakage at doors and elevator shafts. |
| Poultry... | Poultry should be dry picked, dry cooled, and dry packed at temperatures ranging from 30 deg. F. to 35 deg. F. for from 18 to 24 hours, then frozen at 6 deg. F. or below. | The classes and grades should conform to those generally adopted by the wholesale trade, until United States standards are promulgated. | Same as for eggs. | Preferred temperature, 0 deg. F. to 10 deg. F.<br>Admissible temperature, 12 deg. F. to 14 deg. F. | Same as frozen eggs. | Poultry should be so stowed that separate lots may be inspected easily, and protected from injury by heat leakage at doors and elevator shafts. |
| Butter.... | Butter should be placed in cold storage within ten days after it is manufactured. When storage facilities are not available during this period, the product should be held in a temperature below 40 deg. F. | The grades should conform to those generally adopted by the wholesale trade, until United States standards are promulgated. | Same as for eggs. | Temperature, 2 deg. F. or below. | Same as frozen eggs. | Packages of butter should be so stowed as to permit a free air circulation beneath the pile, and so stacked that separate lots may be inspected easily. Cube and box packages should be separated by 1 in. dunnage. |
| Fish..... | Fish should be placed in storage in a fresh condition, as indicated by their physical appearance. | Practically all kinds used for food. | Same as for eggs. | Hard frozen and glazed at temperature of −5 deg. F. or below, and stored at 0 deg. F. or below to 10 deg. F., depending on the kind. For holding less than six months it is admissible to store at 12 deg. F. | Same as frozen eggs. | Fish should be stowed as compactly as possible. |

TABLE 94—*Continued*

| Product | Warehouse Requirements | | Other Considerations | | |
|---|---|---|---|---|---|
| | E—Containers | F—Inspection | Storage period | Shrinkage | Remarks |
| Apples.... | Containers shall be clean, strongly built barrels, boxes or crates, and when packed for market shall be plainly marked with the grade variety and the grower's or packer's name. | All lots of apples should be inspected when received for storage by a qualified inspector. Subsequent inspections of representative packages of all lots should be made at intervals of 15 to 30 days, depending on the variety and condition of the fruit as indicated by previous inspections. | The usual cold storage period for winter varieties of apples is from three to six months, depending upon the variety and condition of the fruit when stored. | The shrinkage in cold storage is from 2 per cent to 5 per cent.<br>In common storage the shrinkage is variable. | Attention is directed to the fact that a delay of one or more weeks between the picking and storing of apples greatly reduces the storage period of the product and results in early deterioration. The successful storage of apples is as much dependent upon the treatment they receive before being placed in cold storage as the conditions under which they are held in storage.<br>See Department Bulletin No. 5877. |
| Potatoes.. | Potatoes may be stored in clean burlap bags, barrels, boxes or crates; or when in bulk they should be stored in ventilated bins. | Potatoes should be inspected by a qualified inspector when received for storage, and again within 30 days. The frequency of the inspections thereafter will depend upon the condition of the potatoes as determined by previous inspections. | The usual storage period is from three to six months, depending upon the section of the country in which the storage is located, the type of the storage house and the condition of the stock. Allowing for a high percentage of deterioration, potatoes may be held in storage for a much longer period. | When potatoes are stored, in containers or in bulk, as specified in Column 6, the shrinkage may amount to about 7 per cent, although it varies greatly. | Potatoes are usually stored in cellars and common storages, but are sometimes held in cold storages.<br>See Farmers' Bulletin No. 847. |
| Potatoes (sweet).. | Sweet potatoes are usually stored in bins, but may be stored satisfactorily in substantial crates, baskets or hampers which permit of a free air circulation. | The potatoes should be thoroughly inspected by a qualified inspector at the time they are put in the storage house, within 15 days after the beginning of the storage period and from 15 to 30 days thereafter. | The safe storage period is about four months. Under the most favorable conditions and good management sweet potatoes may be kept six months. | The shrinkage from loss of moisture is from 8 per cent to 10 per cent in bins, and somewhat higher in packages. An additional shrinkage of 5 per cent should be allowed for decay. | It is recommended that sweet potatoes be not considered properly stored until they have passed through the curing period.<br>See Farmers' Bulletin No. 970. |
| Onions.... | The best containers are slatted crates, although baskets, hampers and bags are used successfully. | Thorough inspection should be made when the onions are placed in storage, and at intervals not exceeding 30 days. The frequency of the inspections thereafter will depend upon the condition at the previous inspection. | The usual storage period of onions with proper ventilation is six months. | The shrinkage should not exceed 10 per cent or 12 per cent. | See Farmers' Bulletin No. 354. |

| | | | | | |
|---|---|---|---|---|---|
| Cabbage | Containers are not generally used. | Cabbage should be inspected at intervals of from 15 to 30 days. | The storage period for cabbage extends from November to April—five or six months. | The shrinkage in cabbage is quite variable. | Stoves should be provided in common storages to prevent freezing in cold periods. See Farmers' Bulletin No. 433. |
| Eggs | Eggs should be packed in clean, odorless, wood cases. Fillers should be of new No. 1 or medium straw or wood pulp board with flats over top and under bottom. Padding must be kiln-dried excelsior, cork shavings or corrugated straw or wood pulp board on top and bottom of each case. No pine excelsior should be used. The cases should be plainly marked with the grade. | Inspection of eggs should be at intervals of from 15 to 30 days, and the storage house should have daily attention from a competent warehouseman skilled in the handling of such structures and commodities. | The storage period for eggs should not exceed 12 months. | The shrinkage depends upon the humidity, and should not be more than 5.5 per cent. Shrinkage should be calculated from net weight of products. | Rooms must be clean and odorless. See Bureau of Chemistry Circular No. 64. |
| Frozen eggs | Thirty-pound tin buckets are most common. The use of smaller tin cans is now increasing, due to wider use of this product. | Inspections of frozen eggs should be made about every 30 days. | No change in composition up to 24 months. After 12 months egg thickens slightly. Whites near top of can may become pink, due to iron under tin. Egg not injured as food thereby. | The shrinkage is not of commercial importance. | See Department Bulletin No. 51. Department Bulletin No. 224. Bureau of Chemistry Circular 98. |
| Poultry | All poultry should be packed in clean, strongly built, odorless boxes, lined with parchment or other suitable paper, and should be plainly marked to indicate the grades and classes. Barrels are still admissible, especially for turkeys, but are less desirable than boxes. | All lots of poultry should be inspected by a qualified inspector when received for storage, and at intervals of 30 days or longer, depending upon the conditions found at the previous inspections. | The storage period for poultry should not exceed 12 months. | The shrinkage varies from 1 per cent to 3 per cent. | Water-cooled or ice-packed poultry should not be stored for long periods. Scalded birds deteriorate more rapidly than dry-picked. Drawn poultry should never be stored. See Bureau of Chemistry Circulars Nos. 64 and 70. |
| Butter | Packages should conform to the regular commercial styles, including 63 lb. tubs, 63 to 78 lb. cubes and standard boxes of 1 lb. prints. | All lots of butter should be inspected by a qualified inspector when received for storage, and at intervals of 30 days or more, depending upon the quality and condition of the lots at previous inspection. | The storage period for butter should not exceed 12 months. | In general, the shrinkage will run from ½ per cent to 1 per cent. | See Bureau of Animal Industry Bulletins Nos. 84 and 148. |
| Fish | Fish are stored in boxes and in bulk. | Inspection of fish should be made at intervals of 30 days or more by a qualified inspector. | The storage period for fish should not exceed 12 months. | The shrinkage is not of commercial importance. | Boxed fish should be reglazed in three to six months. Stacked fish should be spray-glazed every three months or more frequently. See Department Bulletin No. 635. |

The *moth miller* and the *beetle* are killed at a temperature of 40 deg. F., but relatively slowly, whereas at 32 deg. F. they are killed quickly. If they are rolled up in rugs or garments they may be protected for weeks, but the evidence is that no damage is done under the conditions of 32 deg. to 40 deg. F. In fur or rug storage it is better to hang up the garment and to lay the rug out in flat piles, and the temperature must be 40 deg. or lower. The larvæ may survive a temperature of 18 deg. F. for weeks, but usually it may be killed if the temperature is permitted to rise to 45 to 50 degrees for a day or two and then lowered again. As a rule, 25 degrees is the preferred operating temperature.

## EFFECT OF COLD STORAGE ON CLOTHES MOTHS

By E. A. Back and R. T. Cotton, Bureau of Entomology, U. S. Department of Agriculture

Cold storage is the best method of protection against clothes moth damage. Once in cold storage no injury can take place. Cold storage can be depended upon for absolute protection by dealers in furs, carpets and other valuable articles such as stuffed animal heads, blankets, carriage robes, curtains, upholstered furniture, etc. This is true because clothes moth larvæ or worms cannot feed at temperatures below 45 deg. F. A temperature range lower than one of 40 deg. to 42 deg. or 45 deg. F. is unnecessary.

If cold storage will protect, and has the sanction of the United States Department of Agriculture, why is it that warehousing concerns often have complaints lodged against them by patrons who state that living moths' larvæ are found by them in articles several days after removal from cold storage? The presence of living larvæ in such articles can be explained in two ways. First, larvæ may have crawled to the articles after they were removed from storage from other infested material in the home, but this possibility is rather remote if living worms are detected within several days; second, that while cold storage temperatures were sufficient to protect, they were not sufficient to kill. Cold storage concerns should determinine in advance as a matter of policy whether they offer a service of protection against damage for the period of storage, or whether in addition to this they are to guarantee the articles refrigerated to be free from living moths at the time they leave storage.

There is very little exact data on the effect of cold temperature upon the different stages of clothes moths. Some years ago Dr. A. M. Reed, of the Security Storage Company in Washington, D. C., conducted experiments in cooperation with Dr. L. O. Howard, Chief of the Bureau of Entomology, which indicated that larvæ of the webbing clothes moth *tineola biselliella* can survive for a long time a temperature of 18 deg. F. No explanation was given of what a "long time" meant in actual days. It was found, as has been discovered in the case of other insects, that it isn't always the cold alone that kills, but the sudden fluctuations from a cold to a warmer temperature and back to a cold temperature. These experiments, referred to above, resulted in the present recommendation of the Department that to kill clothes moths in storage the infested articles should be refrigerated at 18 deg. F. for several days, then suddenly exposed for a short time to 50 deg. F., and then returned to 18 deg. F., and finally held permanently at about 40–45 deg. F.

If a concern merely guarantees to protect articles during the period of storage, it is sufficient to maintain a temperature of about 40–45 deg. F. At this range the eggs of the moth which are apt to be on the article at time of storage will be killed if the period of storage is prolonged over six weeks. Experiments have shown that at 20–25 deg. F. and at 25–30 deg. F., clothes moth eggs are all killed in about three weeks; at 30–35 deg. F., in 26 days, and at 35–40 deg. F. in about one month. But the older well-grown larvæ are very resistant. While these well-grown larvæ were killed in 67 days when subjected to an even temperature of 20–25 deg. F., and in 93 days at 25–30 deg. F., others held at temperatures ranging from 30–35 deg. F. and 35–40 deg. F. are still alive after over four months.

The ability of well-grown larvæ to withstand long periods of refrigeration at moderate temperatures was demonstrated recently in a Washington plant whose rooms were said to have had a temperature range of 24–48 deg. F., but with the temperature mostly at about 40 deg. F. Larvæ were removed after storage at the end of 6, 8, 10, 12, and 14 months. Storage for 6 to 8 months had no appreciable effect upon larval mortality. After 10 to 12 months' storage, many larvæ died soon after removal, and only a few survived 14 months' storage. Yet these few were thoroughly normal, and upon being placed in a sufficiently warm temperature for feeding, resumed activity and transformed the pupæ and adults as though their life had not been prolonged by an enforced 14-month hibernating period.

These facts are important for they explain why patrons have at times found living robust larvæ in articles several days after removal from 4 to 5 months in cold storage. If a fairly even temperature around 40 deg. F. has been maintained, such a discovery is to be expected and is no reflection upon the storage firm. Experiments under way will determine definitely the effect of various temperature ranges upon not only the eggs and the well-grown larvæ, but upon the larvæ of different ages.

The fur storage room is not opened to much of any extent after the goods are placed in storage until after the summer is over and the refrigeration required is nearly all "heat leakage." Such storage rooms require no windows of any sort, and in consequence no correction has to be made for window infiltration or leakage. Electric lights are used, of course, but the amount of heat generated by their use is very small after the goods are once stored. Only one door is usually provided, and that is of the usual cold storage construction and is generally tight and of the same insulating value as the rest of the room. The insulation is usually of 5 to 6-in. corkboard, erected in thicknesses of 2 in. with the joints broken and put up in cement mortar or asphalt. A half-inch cement mortar finish is given the inside of the walls and the ceiling.

In providing refrigeration, piping in the rooms is almost never used, except, perhaps, in the very small rooms. If piping is used it should be very carefully protected so that the frostation on the pipes could never drip on to the goods. There is danger also from the pipes should leaks occur, both from the brine and the ammonia. It is not known whether carbon dioxide is innocuous to furs and fabrics, but the understanding is that no appreciable harm occurs. The preferred method of cooling is by the use of a bunker room. With small rooms the air may be admitted

in one or two places, but the larger rooms need distributing ducts for the supply and the return. Such ducts may, of course, be designed for velocities as high as 2100 ft. per min., but as a rule lower velocities are used—from 1200 to 1600 being usual. Refrigeration is different from heating and ventilation as regards the use of fans. The fan not only uses power to drive the air, but the heat equivalent of the work done must be neutralized by the use of the refrigeration machine. Excess power supplied then is a double expense in the case of refrigerating plants, and in fact there is required a $\frac{1}{5}$ ton of refrigeration for every horse power supplied to the fan.

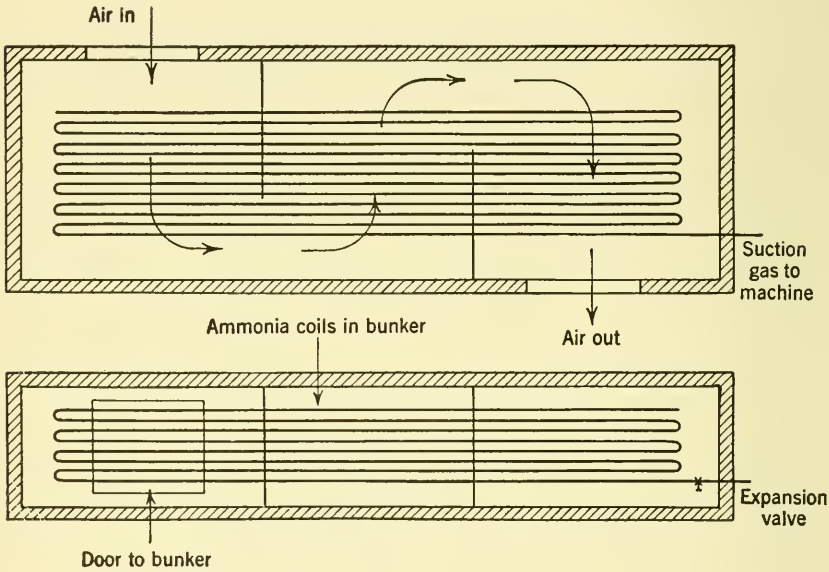



Fig. 283.—Simple Bunker Room Coil for Direct Expansion of Ammonia.

The *bunker* is usually designed (as in the case of the chocolate dipping spray chamber) for 550 ft. per min. air velocity, and the bunker is usually arranged for three passes of the air, and in some of the constructions the baffles are used to catch the condensate and guide it to the waste pipe. Where the size of the plant warrants it, there is no reason why the brine spray chamber cannot be used, with eliminators as in the case of the cooling of air. In fact there are now on the market small spray cabinets which would meet any requirement. As air infiltration is very small, and the commodities do not dry out as do vegetables, fruits and meats, there will be little moisture condensed and therefore very little trouble from diluting of the brine.

The figure (Fig. 283) shows a simple pipe-coil bunker room for ammonia. Ammonia piping is now designed for little or no return bends or other fittings. Instead the pipes are bent and the separate pipes are welded, usually by the electric method. A number of such pipes are welded together (as many as can be handled without difficulty during erection) and similar units are joined together by means of the usual tongue and groove flanges. Or, sometimes the job is made up at the point of erection by the use of thermit welding of different coils, and in this manner a complete welded system of piping is obtained, but thermit welding is much more costly.

TABLE 95

Carload Shipments to Principal Cities

Figures containted in Information Bulletin No. 272, of the Perishable Freight Conservation Bureau, American Association of Ice and Refrigeration, give some idea of the large amount of fruits and vegetables consumed in the principal cities of the United States. While it is not known just what proportion of these cars were refrigerated, it is presumed that a very large percentage of them, owing to the character of the product transported, were under refrigeration. The number of carloads unloaded at the various cities are as follows:

| Unloaded at | Carloads |
|---|---|
| Boston | 30,528 |
| Chicago | 48,222 |
| Cleveland | 12,389 |
| Cincinnati | 10,573 |
| Detroit | 11,969 |
| Kansas City | 7,601 |
| Minneapolis | 3,797 |
| New York | 94,194 |
| Philadelphia | 28,559 |
| Pittsburgh | 20,145 |
| St. Louis | 12,115 |
| St. Paul | 1,891 |
| Washington, D. C. | 5,014 |

In the case of certain commodities the records and reports did not begin until June 1, 1923.

From the foregoing it is evident that the refrigerating load is a small one. It is almost entirely a leakage load, and therefore it depends on the insulation provided. With 5- or 6-in. corkboard the volume of space per ton of refrigeration is about 3000 cu. ft. But this is a case where the load can be calculated with great precision taking the maximum

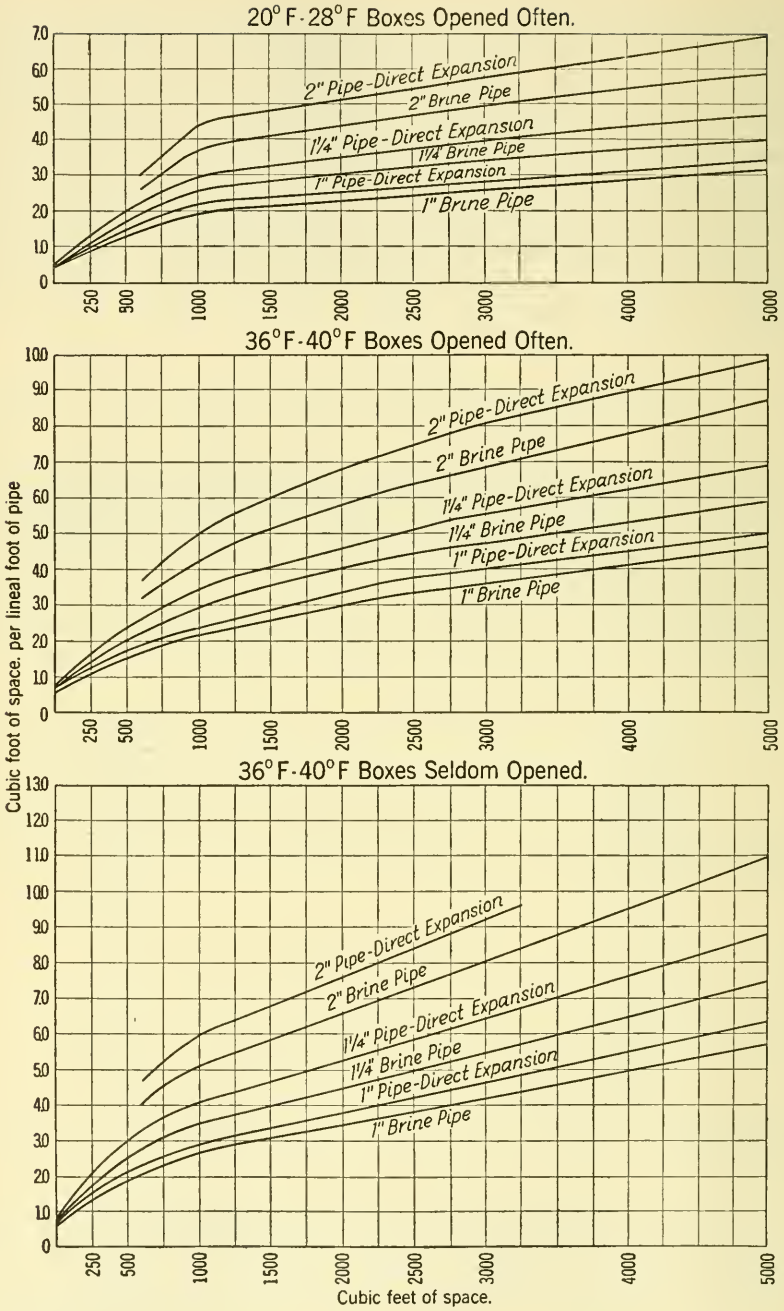



Fig. 284.—Piping for Small Cold Storage Boxes.

sustained temperature usual in the locality where the installation is to go, and designing the plant for not more than 12 hours' operation, thereby giving a factor of safety if more severe conditions prevail.

**Ozone.**—Ozone when used in cold storage work is in amounts less than one volume in one million, which concentration has been found to be non-germicidal. Ozone cannot be made to kill flourishing fungi when used in such small concentrations, but it is quite possible to prevent the *development* of *fungi* by its use. Ozone[13] will inhibit the development of spores but will not destroy mold growth. "Ozone is a prophylactic and not a remedial agent. It has been found by experiment that air with a mild concentration of ozone has prevented mold growth in egg storage carrying a relative humidity of 90 per cent. In egg and bacon rooms with very high humidity, molds have been effectively inhibited. Equally good results have been obtained in general cold storage practice." Mr. Hartman says that all odorous substances give off gases or finely divided particles, and when oxidized they become non-odoriferous. Ozone has been found satisfactory for freeing the atmosphere of these gases, and thus preventing taste transfer or modification. Onions can be stored in the same room with fruits and vegetables without transfer of taste, and it has been found possible to combat mold formation on grapes and other fruits. In storage of such commodities as cheese, the air in the room has been maintained fresh and free from odors, whereas ozone itself has never caused deleterious effects on the goods stored.

**Calculating the Refrigerating Load.**—The refrigerating load is very hard to calculate at times with accuracy, and so it has become the custom of refrigerating engineers to use the method of the rule of thumb. This is because there are so many variable factors which may affect the machine capacity or the amount of piping in the room that it has been found desirable to install the refrigeration or the piping surface from experience with other and similar jobs where satisfactory results have been obtained. For example, Fig. 284 gives a set of curves of such an origin for different sizes of boxes and operating temperatures and conditions, such as whether the boxes are to be "opened frequently" or "seldom." It is quite evident that such curves have a value, but the box that would be satisfactory in New York need not necessarily be satisfactory in Alabama. These curves then are for the practice on which they were based, and similar curves are sometimes marked for "good insulation" and for "poor insulation." However, the small cold storage box, say of 5000 cu. ft. interior contents and

[13] F. E. Hartman, American Society of Refrigerating Engineers Journal, November, 1924.

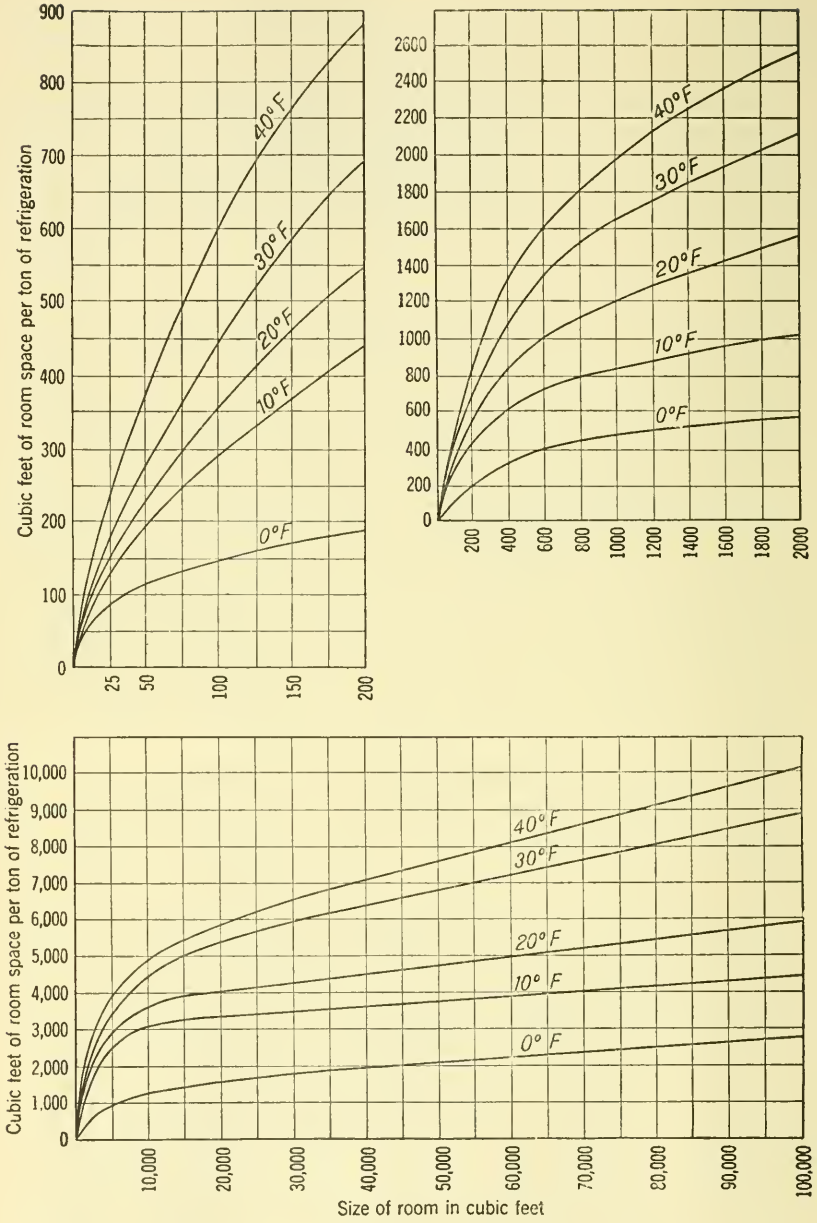



Fig. 285.—Piping for Cold Storage Rooms with Usual Insulation.

under, is practically impossible of accurate calculation as regards the load to be encountered. Such boxes are carelessly cared for, the doors are opened perniciously and often left open needlessly. The small box

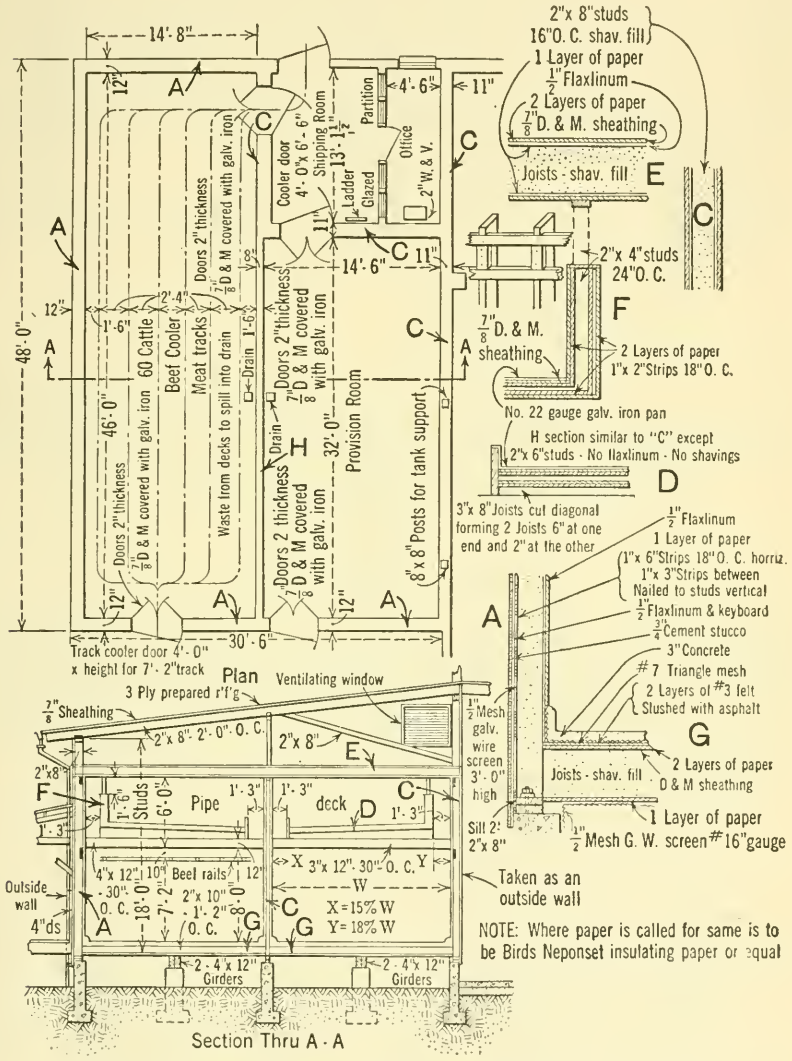



FIG. 286.—Construction of Cold Storage Rooms [for Problem].

has (unless built for some particular purpose) no well-defined load, and so the surface installed in such boxes must be liberal.

The piping for the large cold storage room (Fig. 286), on the contrary,

is more easily calculated. Such a room is likely to be of permanent building construction, and the insulation will be comparable with the rest of the building. The usual manner of calculation is to make an estimate of the heat leakage under the assumed conditions and then the refrigeration required to cool the "live load." The following illustrative example will serve as a guide, and such construction is representative of the temporary buildings constructed by the United States Government for the army cantonments. The calculation is identical with those used in Chapter VI in solving for heat leakage.

*Cold Storage Problem.*—The cold storage rooms are shown in Fig. 286. The live load will consist of the following commodities received daily:

1 car of beef (a car holds 40 beeves at 600 lb.).
20 barrels of apples.
50 barrels of potatoes.
20 cases of eggs (of 60 lb.).
10 tubs of butter.

The beef and butter arrive at 60 deg. F. The apples, potatoes and eggs will be taken as arriving at 80 deg. F. The operating suction pressure will be taken at 30 lb. gage.

Take the specific heat of beef = 0.77 (average)
apples = 0.92
potatoes = 0.80
butter = 0.60
eggs = 0.76

The outside temperature will be taken at 90 deg. F., and the temperature in the beef and the provision rooms will be held at 33 degrees. The first problem will be the calculation of the heat leakage.

*Wall "A".*—This is composed of $\frac{7}{8}$-in. sheathing, $\frac{1}{2}$-in. flaxlinum, 9 in. of shavings, $\frac{3}{4}$ in. stucco, and $\frac{1}{2}$ in. flaxlinum (neglect insulation effect of building paper).

Using the formula for heat leakage

$$u = \frac{1}{\frac{1}{k_1} + \frac{1}{k_2} + \frac{T_1}{C_1} + \frac{T_2}{C_2} + \frac{T_3}{C_3} + \frac{T_4}{C_4}, \text{ etc.}}.$$

using the values for this construction,

$$u = \frac{1}{\frac{1}{1.4} + \frac{1}{4.2} + \frac{1}{1.0} + \frac{(\frac{1}{2} + \frac{1}{2}) \times 24}{7.9} + \frac{9 \times 24}{10} + \frac{0.75}{8.0}} = 0.03745.$$

*Ceiling "E."*—2-$\frac{7}{8}$ in. sheathing, 2-$\frac{1}{2}$ in. flaxlinum, and 8 in. shavings.

$$u = \frac{1}{\frac{1}{1.4} + \frac{1}{4.2} + \frac{2}{1} + \frac{(\frac{1}{2} + \frac{1}{2}) \times 24}{7.9} + \frac{8 \times 24}{10}} = 0.0397.$$

*Wall "C."*—This is the same as the *E* construction.

$u = 0.0397.$

*Floor "G."*

$$u = \frac{1}{\frac{1}{0.93} + \frac{1}{4.2} + \frac{3}{1} + \frac{10 \times 24}{10} + \frac{3.0}{8.0}} = 0.03486.$$

Heat Leakage (meat room):

| | | |
|---|---|---|
| Sides and 2 ends | $0.03745 \times (644 + 205 + 205)(90 - 33)$ | = 2250 B.t.u. |
| Floor | $0.03486 \times 674 \times (90 - 33)$ | = 1340 B.t.u. |
| Ceiling | $0.0397 \times 674 \times (90 - 33)$ | = 1526 B.t.u. |
| Side | $0.0397 \times 448 \times (33 - 33)$ | = 0 B.t.u. |
| Side (shipping room) | $0.0397 \times 184 \times (90 - 33)$ | = 416 B.t.u. |
| | Total | = 5532 B.t.u. |
| | Add 20 per cent | 1106 |
| | | 6638 B.t.u. per hr. |

Heat Leakage (provision room):

| | | |
|---|---|---|
| Side and one end | $0.0397 \times (448 + 203)(90 - 33)$ | = 1474 B.t.u. |
| Floor | $0.03486 \times (464)(90 - 33)$ | = 922 B.t.u. |
| Ceiling | $0.0397 \times 464 \times (90 - 33)$ | = 1050 B.t.u. |
| One end | $0.03745 \times 203 \times (90 - 33)$ | = 433 B.t.u. |
| | Total | = 3879 B.t.u. |
| | Add 20 per cent | 776 B.t.u. |
| | | 4655 B.t.u. per hr. |

The second problem is the calculation of the live load. The assumption is that the temperature of the commodities is reduced to 33 deg. F. in 24 hours.

| Live Load | Specific Heat | Degrees Cooling | Total Weight | Total Refrigeration, B.t.u. |
|---|---|---|---|---|
| Beef | 0.77 | 27 | 40×600 | 499,000 |
| Apples | 0.92 | 47 | 20×150 | 129,500 |
| Potatoes | 0.80 | 47 | 50×180 | 338,500 |
| Butter | 0.60 | 27 | 10×63 | 10,200 |
| Eggs | 0.76 | 47 | 20×60 | 42,900 |
| Total | | | | 521,100 |

Total Load, meat room, per hour = 499,000 ÷ 24 + 6638 = 27,430 B.t.u. per hour.
Provision room = 521,100 ÷ 24 + 4655 = 26,360 B.t.u. per hour.

Piping required:

Meat room $27{,}430 \div [1.6 \times (33.0 - 16.6)] = 1045$ sq. ft.
$= 2405$ lin. ft. $1\frac{1}{4}$-in. pipe.
Provision room $26{,}360 \div [1.6 \times (33.0 - 16.6)] = 1002$ sq. ft.
$= 2320$ lin. ft. $1\frac{1}{4}$-in. pipe.

The arrangement of the piping will be as indicated by Fig. 287, which shows the expansion valve between two stop valves and the liquid feed at the bottom of the coils. An allowance of 20 per cent excess heat leakage is made for the opening of doors, the heating effects of lights and of workmen in the room, etc. It is assumed that the live load will be reduced to such an extent that the temperature of the commodities will be taken at 33 deg. F. at the end of 24 hours, and that the coefficient

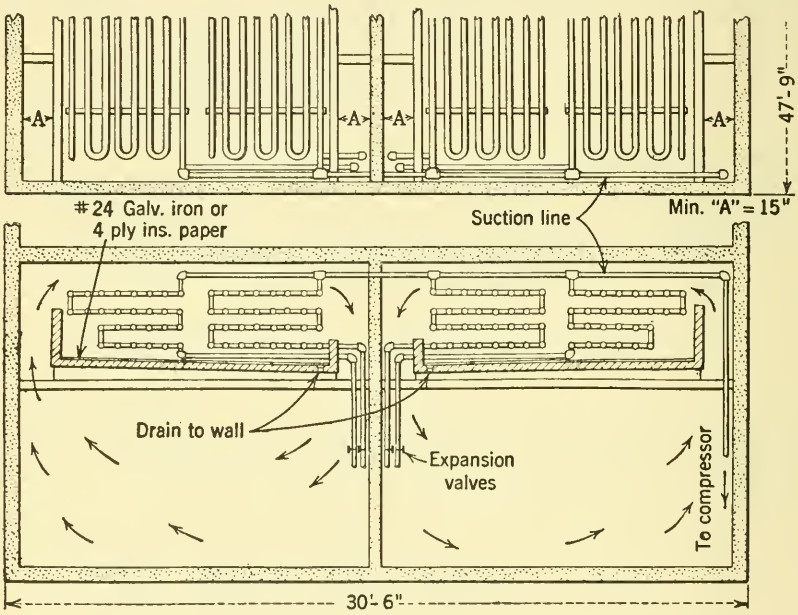
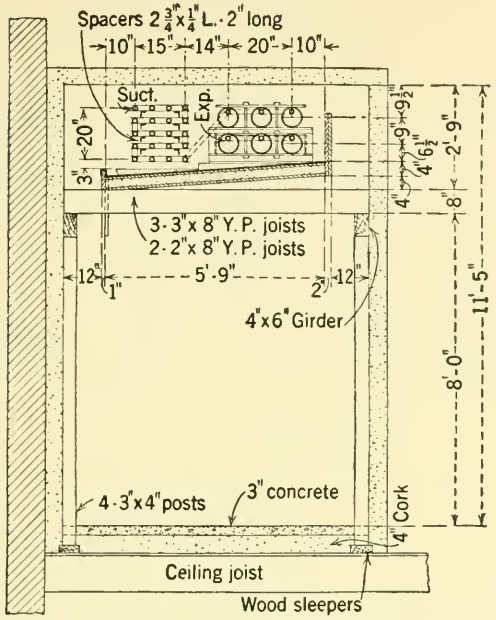



Fig. 287.

of heat transfer ($k$) of the pipes is equal to 1.6 B.t.u.[14] The tonnage on the machine becomes $\frac{27{,}430 + 26{,}360}{12{,}000} = 4.47$ tons. The size of the suction line is found (at 155 lb. condenser pressure and 30 lb. suction pressure, and with a velocity of the gas in the return suction line of 4000 ft. per minute, by the following method): Volume of gas $\frac{200}{616.8 - 139.1} \times 6.35 \times 4.47 = 11.9$ cu. ft. per min. The cross-

[14] The value of 1.6 for $k$ is low, but is conservative on account of possible frostation of the pipes and careless operation.




Cross Section C-D

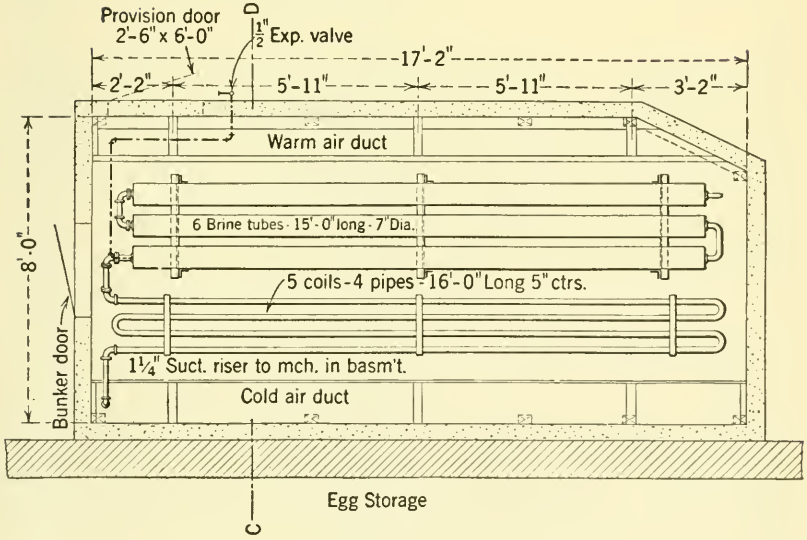



FIG. 288.—Brine Tubes and Ammonia Piping in Pipe Deck.

## TABLE 96

### Composition and Specific Heat of Food Products and Storage Temperatures

| Product | Temperature Carried | Composition, Per Cent | | Specific Heat above Freezing | Specific Heat below Freezing | Latent Heat of Freezing |
|---|---|---|---|---|---|---|
| | | Water | Solids | | | |
| Lean beef | 30 | 72.00 | 28.00 | .77 | 0.41 | 104 |
| Fat pork | 30 | 39.00 | 61.00 | .51 | .30 | 56 |
| Eggs | 30 | 70.00 | 30.00 | .76 | .40 | 101 |
| Potatoes | 34 | 74.00 | 26.00 | .80 | .42 | 107 |
| Cabbage | 33 | 91.00 | 9.00 | .93 | .48 | 131 |
| Carrots | 33 | 83.00 | 17.00 | .87 | .45 | 120 |
| Cream | 33 | 59.25 | 30.75 | .68 | .38 | 85 |
| Milk | 35 | 87.50 | 12.50 | .90 | .47 | 126 |
| Oysters | 35 | 80.38 | 19.62 | .84 | .44 | 116 |
| Whitefish | 15 | 78.00 | 22.00 | .82 | .43 | 112 |
| Chickens | 28 | 73.70 | 26.30 | .80 | .42 | 106 |
| Ice | 23 | 100.00 | 0.00 | 1.00 | .504 | 144 |

### Approximate Cold Storage Temperatures

| Article | Temperature, Degrees F. | Article | Temperature, Degrees F. |
|---|---|---|---|
| Apples | 30 | Huckleberries, frozen | 20 |
| Asparagus | 33 | Ice | 28 |
| Bananas | 55 | Ice cream, short carry | 15 |
| Beans, fresh | 32 | Lemons, short carry | 50 |
| Beans, dried | 45 | Lemons, long carry | 38 |
| Beef, fresh, short carry | 35 | Lambs | 32 |
| Beef, fresh, long carry | 30 | Lard | 40 |
| Beef, dried | 40 | Livers | 20 |
| Beer, in barrels | 32 | Maple syrup and sugar | 45 |
| Beer, in bottles | 45 | Meats canned | 40 |
| Berries, fresh, short carry | 40 | Meats, salt, after curing | 43 |
| Buckwheat flour | 42 | Milk, short carry | 35 |
| Butter | 14 | Nursery stock | 30 |
| Butterine | 20 | Nuts in shell | 40 |
| Cabbage | 33 | Oatmeal | 42 |
| Cantaloups, short carry | 40 | Oils | 45 |
| Cantaloups, long carry | 33 | Oleomargarine | 20 |
| Carrots | 33 | Onions | 32 |
| Celery | 32 | Oranges, short carry | 50 |
| Cheese, long carry | 35 | Oranges, long carry | 34 |
| Chestnuts | 34 | Oxtails | 30 |
| Chocolate dipping room | 65 | Oysters in shell | 43 |
| Cider | 32 | Oysters in tubs | 35 |
| Cigars | 42 | Parsnips | 32 |
| Corn, dried | 45 | Peaches, short carry | 50 |
| Cornmeal | 42 | Pears | 33 |
| Cranberries | 33 | Peas, dried | 45 |
| Cream, short carry | 33 | Plums | 32 |
| Cucumbers | 38 | Potatoes | 34 |
| Currants, short carry | 32 | Poultry, dressed, iced | 30 |
| Dates | 55 | Poultry, short carry | 28 |
| Eggs | 30 | Poultry, after frozen | 10 |
| Figs | 55 | Poultry to freeze | 0 |
| Fish, not frozen, short carry | 28 | Raisins | 55 |
| Fish, fresh water, frozen | 18 | Ribs, not brined | 20 |
| Fish, salt water, not frozen | 15 | Salt meat curing room | 32 |
| Fish, to freeze | 5 | Sardines, canned | 40 |
| Fish, dried | 40 | Sauerkraut | 38 |
| Flowers, cut | 36 | Sausage casings | 20 |
| Fruits, canned | 40 | Scallops, after frozen | 16 |
| Fruits, dried | 40 | Sheep | 32 |
| Furs | 28 | Shoulders, not brined | 20 |
| Game, short carry | 28 | Sugar | 45 |
| Game, after frozen | 10 | Syrup | 45 |
| Game, to freeze | 0 | Tenderloins | 33 |
| Ginger ale | 36 | Tobacco | 42 |
| Grapes | 36 | Tomatoes, ripe | 42 |
| Hams, not brined | 20 | Watermelons, short carry | 40 |
| Hogs | 30 | Wheat flour | 42 |
| Honey | 45 | Wines | 50 |
| Hops | 32 | Woolens | 28 |

TABLE 96—*Continued*

COLD STORAGE PRACTICE

| Temperature of Storage, Degrees F. | Humidity, Per Cent | Allowances |
|---|---|---|
| | | 1 bbl. = 31½ gal. of 231 cu. in. each. |
| | | 1 bushel = 1.245 cu. ft. = 2150.4 cu. in. |
| 35 | 80 to 90 | 1 bbl. potatoes = 5 cu. ft. = 180 lb. = 2½ bushels. |
| 32 to 34 | 70 to 80 | 1 box cheese = 60 lb. = 2 cu. ft. |
| 30 to 31 | 75 | 1 case eggs = 30 doz. = 50 − 70 lb. = 2¼ cu. ft. = 12 × 13 × 25. |
| 2 or lower | 70 | 1 tub butter = 63 lb. = 1½ or 2 cu. ft. (piles not over 6 ft.). |
| 30 to 35 | 85 | 1 bbl. apples = 150 lb., 5 cu. ft. = 2½ bushels. |
| 34 | 80 to 90 | 1 crate celery = 10 cu. ft. = 140 lb. = 24 × 24 × 30 in. (8 ft. high). |
| 34 | 80 to 90 | 1 bbl. vegetables = 5 cu. ft., piles 5 ft. high. |
| 35 | 80 | 1 box oranges (packed on end) = 4 cu. ft., 70 lb. = boxes 15×15×30. |

Aisles—Allow ½ for eggs, celery and oranges.
Allow ⅓ for potatoes and vegetables.
Allow ¼ for mutton, pork, poultry (4 ft. high).
Specific gravity of boned beef = 1073 = 67.2 lb. per cu. ft.

sectional area of the suction pipe $= \frac{11.9 \times 144}{4000} = 0.428$ sq. in., and as the ¾-in. pipe has a transverse area of 0.533 and the ½-in. pipe has one of 0.304 sq. in. the ¾-in. pipe would be the one to be chosen for satisfactory results unless the connections on the compressor are for a larger pipe and it seems desirable to make the entire suction pipe of this size.

**Hold Over Tanks and Pipes.**—Small plants find it uneconomical to operate at times more than 10 to 12 hours per day. Such plants, if considerable temperature rise is to be prevented, require under these conditions either brine operation and a brine tank of sufficient storage capacity to carry the load during periods of shut-down of the compressor, or must provide some auxiliary means to absorb the heat leakage during this part of the day. The accepted method is to use brine storage (of congealing) tank or tubes. In either case a certain amount of brine is cooled during the normal operation of the compressor, at the same time that the room is cooled by the usual piping, and piping arrangements, by means of direct expansion piping submerged by the brine, are so located as to take the place of the direct expansion piping. This scheme is seldom better than a compromise, but there is some advantage in its use. At times 6-in., 7-in. or 8-in. pipe can be used to better advantage than a tank, in which case the pipes are filled with brine. If in the previous problem the provision room was to be controlled during 12 hours by the brine pipe the leakage would be (for 70 deg. F. average temperature outside) 2520 B.t.u. per hour, and in 12 hours this would accumulate to 30,240 B.t.u. If the brine is cooled 20 degrees below the temperature of

the room (that is to 13 deg. F.), and the brine rises 15 degrees during the 12 hours, the requred amount of brine becomes

$$30{,}240 = \text{Weight} \times 0.83 \times 15.$$

$$\text{Weight of brine} = 30{,}240 \div 12.45 = 2430 \text{ lb.},$$

and using a specific gravity of brine of 1.16,

$$2430 \div (1.16 \times 62.3) = 33.7 \text{ cu. ft.}$$

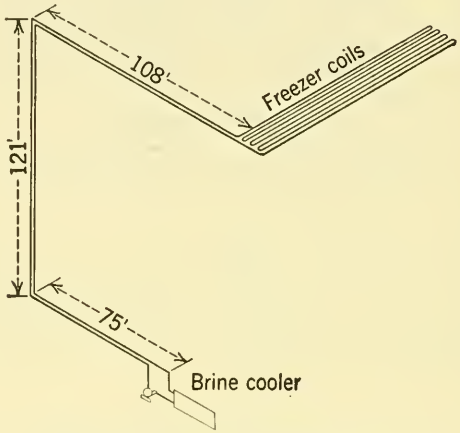



Fig. 289.—Problem in Brine Piping.

TABLE 97

Refrigerated Space—Summary by Class of Business, October 1, 1925

| Class of Business | Number of Concerns | Space, in Cubic Feet, Held at Temperatures of | | | | Total Space |
|---|---|---|---|---|---|---|
| | | 10 Deg. F. and below | 11 Deg. to 29 Deg. F. inclusive | 30 Deg. to 44 Deg. F. inclusive | 45 Deg. F. and above | |
| Public cold storage | 416 | 41,311,130 | 20,196,430 | 175,244,429 | 5,811,924 | 242,563,913 |
| Private cold storage | 264 | 3,288,072 | 5,313,301 | 14,291,508 | 1,172,946 | 24,065,827 |
| Combined public and private cold storage | 212 | 8,450,311 | 11,173,425 | 36,383,333 | 1,027,711 | 57,034,780 |
| Meat-packing establishments | 397 | 11,530,051 | 16,458,902 | 195,026,108 | 23,150,561 | 246,165,622 |
| Meat-packing establishments doing a public cold-storage business | 34 | 4,142,218 | 6,272,375 | 41,067,020 | 5,355,995 | 56,837,608 |
| Total refrigerated space | 1323 | 68,721,782 | 59,414,433 | 462,012,398 | 36,519,137 | 626,667,750 |

The number of pipes for this amount of hold-over brine can be calculated when the other details are known. If 8-in. pipe is used, then 2.88 ft. are required to contain one cubic foot, and the total linear feet required will be 97.1.

**Friction Head in Brine Pipes.**—Considering the importance of brine in refrigeration it is surprising how little information on the friction head

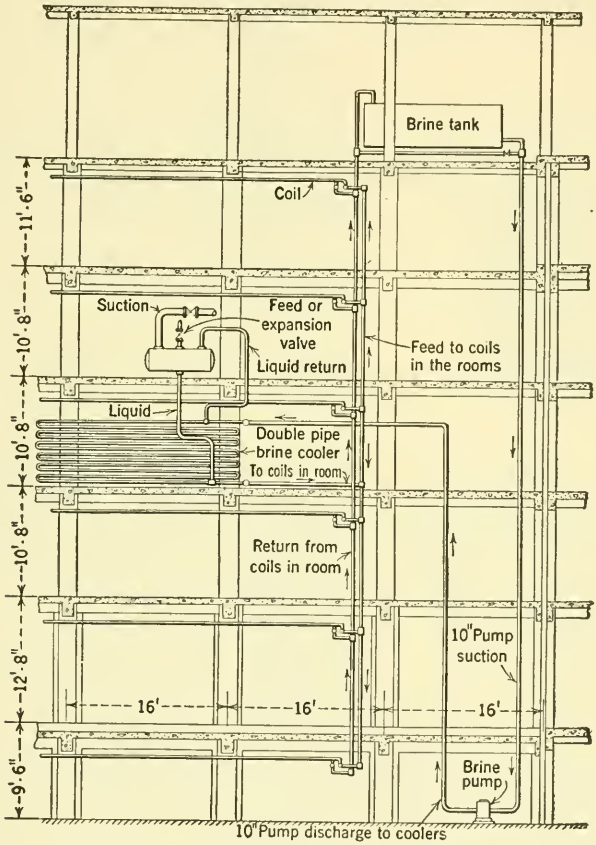



FIG. 290.—Typical Installations—Brine.

loss due to the flow of brine in pipes has so far been collected. The only information on the subject is that of A. H. Gibson, of the University College, Dundee,[15] in which it is shown that the coefficient of friction head increases both with the specific gravity and with a decrease of temperature. The results of his experiments are shown in Fig. 181,

[15] Professor A. H. Gibson, Proceedings of the Inst. of Mech. Eng., London, Feb., 1914. See Chapter VIII for details on this paper.

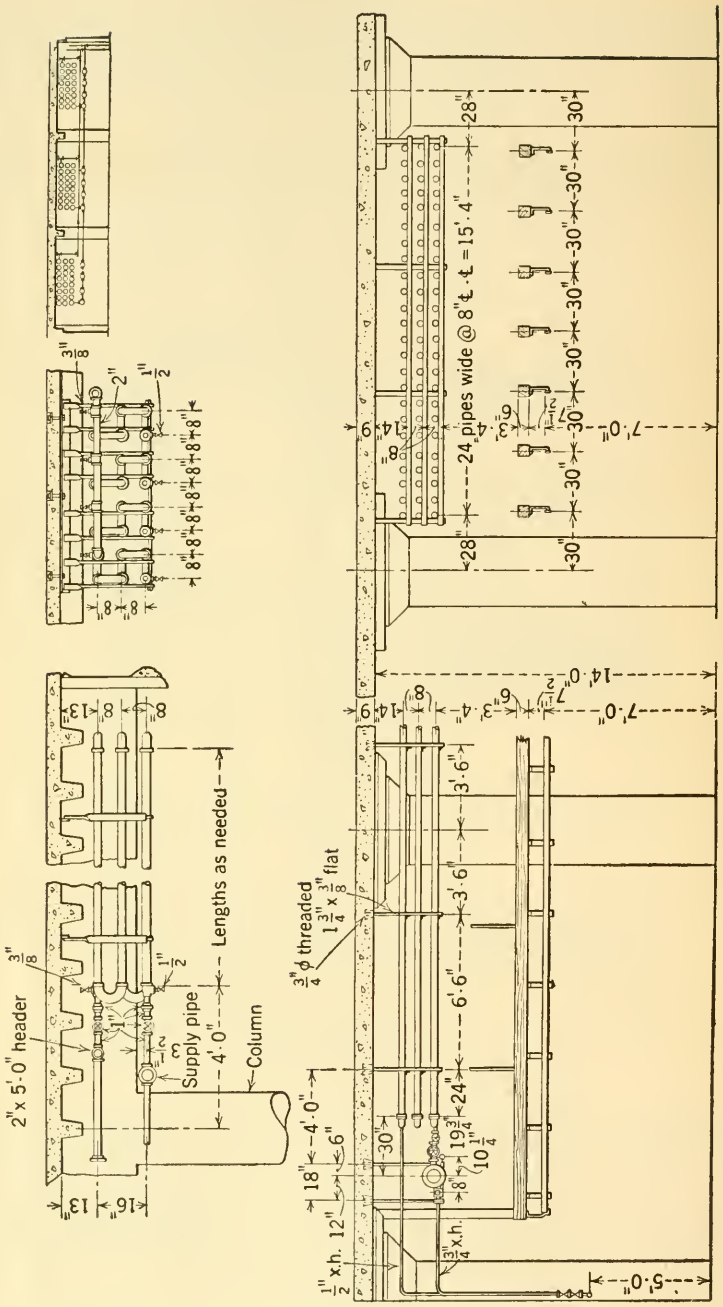



FIG. 291.—Typical Installations—Ammonia.

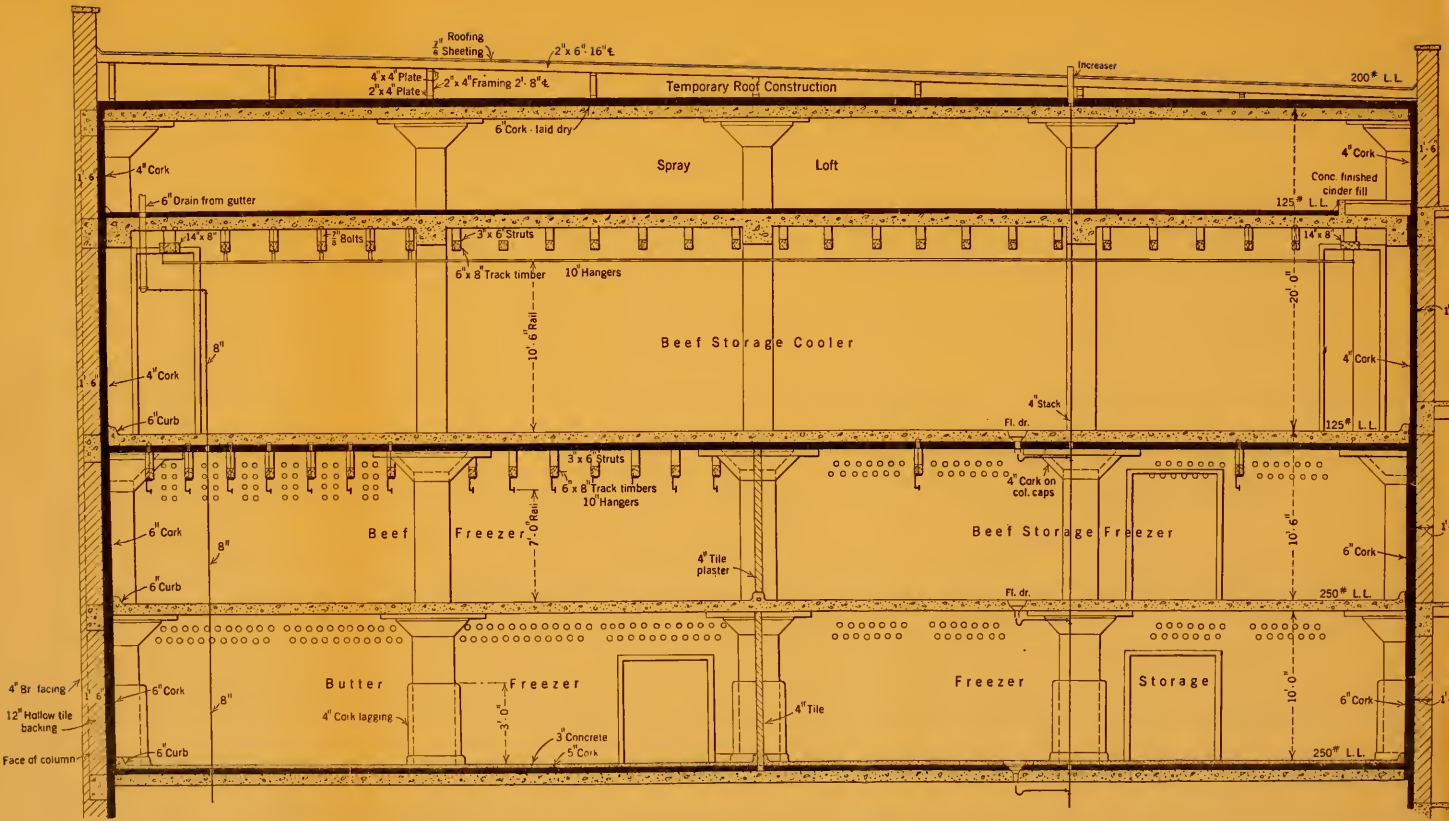



FIG. 292.—Typical Installations—Packing House.

*To face page 402*

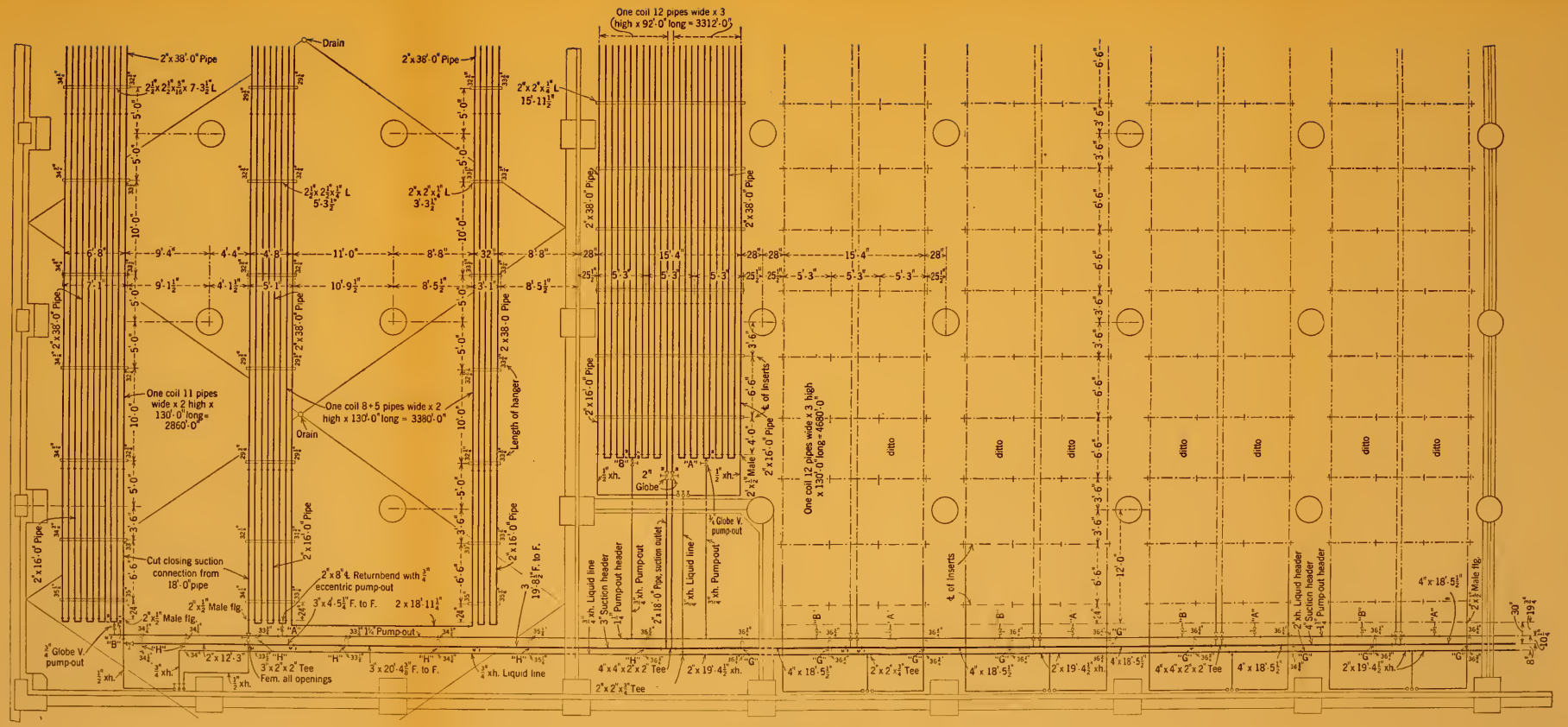

One coil 12 pipes wide x 3 high x 92'-0" long = 3312'-0"
2" x 38'-0" Pipe
Drain
2" x 38'-0" Pipe
2" x 2" x 1/4" L 15'-11 1/2"
2 1/2" x 2 1/2" x 5/16" x 7'-3 1/2" L
2 1/2" x 2 1/2" x 1/4" L 5'-3 1/2"
2" x 2" x 1/4" L 3'-3 1/2"
2" x 38'-0" Pipe
2" x 38'-0" Pipe
2" x 38'-0" Pipe
2" x 38'-0" Pipe
2" x 38'-0" Pipe
2" x 16'-0" Pipe
2" x 16'-0" Pipe
2" x 16'-0" Pipe
2" x 16'-0" Pipe
2" x 16'-0" Pipe
One coil 11 pipes wide x 2 high x 130'-0" long = 2860'-0"
One coil 8+5 pipes wide x 2 high x 130'-0" long = 3380'-0"
Drain
Length of hanger
Cut closing suction connection from 18'-0" pipe
2" x 8" ℄ Returnbend with 1/4" eccentric pump-out
3" x 4'-5 1/4" F. to F.
2 x 18'-11 1/4"
2" x 1" Male flg.
2" x 1" Male flg.
Globe V. pump-out
1/2" xh.
3/4" xh.
2" x 12'-3"
3" x 2" x 2" Tee Fem. all openings
3" x 20'-4 5/8" F. to F.
3/4" xh. Liquid line
2" x 19'-6 1/2" F. to F.
3 1/4" Pump-out
"A" 1" xh.
"B" 1/2" xh.
2" Globe
1/4 Globe V. pump-out
3/4" xh. Liquid line
3" Suction header
1 1/4" Pump-out header
3/4" xh. Pump-out
2" x 18'-0" Pipe, suction outlet
3/4" xh. Liquid line
3/4" xh. Pump-out
4" x 4" x 2" x 2" Tee
2" x 19'-4 1/2" xh.
2" x 2" x 1/4" Tee
One coil 12 pipes wide x 3 high x 130'-0" long = 4680'-0"
℄ of Inserts
2 1/2" Male ℄ of Inserts
2" x 16'-0" Pipe
ditto
ditto
ditto
ditto
ditto
ditto
ditto
ditto
12'-0"
4" x 18'-5 1/2"
2" x 2" x 1/4" Tee
4" x 18'-5 1/2"
2" x 19'-4 1/2" xh.
4" x 18'-5 1/2"
4" x 4" x 2" x 2" Tee
4" x 18'-5 1/2"
2" x 19'-4 1/2" xh.
2" Liquid header
4" Suction header
1 1/4" Pump-out header
4" x 18'-5 1/2"
2 1/2" Male flg.
30"
19 1/2"
10 1/2"
8 1/2"

Fig. 293.—Typical Piping—Ammonia.

*To face page 463.*

which give *multiplying factors* (with reference to the friction head of water) to be used in conjunction with Fig. 180. To use these figures the friction head will be found as if the fluid flowing were water and the proper multiplying factor must be found from Fig. 181. The use of the tables will be shown in the following problem.

**Problem.**—A brine coil of 10,000 lin. ft. of 2-in. pipe is located on the 10th floor of a warehouse. The brine circuit consists of the following: The refrigerating coils, of 2-in. pipe, 10,000 lin. ft. in 10 coils; distance from the brine cooling coils to riser, 108 lin. ft. 3-in. pipe; the riser, 121 lin. ft. 3-in. pipe; riser to brine cooler and pump, 75 lin. ft. 3-in. pipe; total pipe in the circuit, 10 coils of 2 in. each 1000 ft. and 304 ft. of 3-in. pipe.

The freezer room is to be maintained at 32 deg. F. and the brine will enter at 18 degrees and will leave at 22 degrees. The coefficient of heat transfer will be taken at 1.7 B.t.u. for the piping. The amount of brine to be circulated, the friction head loss and the horse power of the pump assuming an overall efficiency of 50 per cent is desired. The arrangement of the piping is indicated in Fig. 290.

The amount of brine to be circulated may be found by the amount of cooling surface. Using a value of $k = 1.7$ B.t.u., the total refrigeration becomes $(10{,}000 \div 1.6) \times 1.7 \times \left(32 - \frac{22 + 18}{2}\right) = 127{,}400$ B.t.u. per hour where the factor 1.6 is the number of feet of 2-in. pipe per one sq. ft. of outside surface.

$$127{,}400 = \text{wt. brine} \times 0.76 \times 4.$$

$$\begin{aligned} \text{Wt.} &= 127{,}400 \div (0.76 \times 4) = 41{,}900 \text{ lb.} \\ &= 41{,}900 \div (62.4 \times 1.15) = 584 \text{ cu. ft. per hour} \\ &= 72.7 \text{ gal. per minute.} \end{aligned}$$

From the chart it will be seen that at a velocity of 3.4 ft. per sec. a 3-in. pipe will have a pressure drop per 100 ft. of pipe of 0.72 lb. (for water) and the friction head loss in the cooling coils (7.3 gal. per min. in 2-in. pipe) will be 0.072 lb. per 100 ft. The total friction head loss is then,

$$0.072 \times 1.3 \times 10 = 0.935 \text{ lb.}$$

$$\frac{(2 \times 108 + 2 \times 121 + 2 \times 75)}{100} \times 0.72 \times 1.3 = 5.69$$

$$\begin{aligned} \text{Total} &= 6.63 \text{ lb.} \\ \text{Add 20 per cent for fittings, etc.} &= 1.33 \\ \text{Total} &= 8.0 \text{ lb.} \\ \text{Work performed} = (41{,}900 \div 60) \times 16.0 &= 11{,}200 \text{ ft.-lb.} \\ 11{,}200 \div 33{,}000 &= 0.340 \text{ hp. for circulating the brine.} \\ &= 0.680 \text{ hp. of the pump assuming 50 per cent efficiency.} \end{aligned}$$

This work performed is under the conditions of a balanced system for the brine. Certain types and designs are shown in Figs. 290 to 293. When the open brine design is used with sprays, the head on the pump is the actual lift plus the friction and the velocity heads. The work performed is then much greater.

## THE PACKING HOUSE

**The Extent of the Industry.**—According to the Bureau of Agriculture Economics for October, 1921, the total refrigerating space in the packing plants in the United States amounts to 258,549,000 cu. ft. Of the amount, which includes 443 plants, 15,400,000 cu. ft. are held at 10 deg. F. or lower; 18,300,000 are held at temperatures between 11 and 29 deg. F., and 200,000,000 cu. ft. are held at temperatures ranging from 30 to 44 deg. F. The total refrigerating space, according to the same authority, is 543,600,000 cu. ft. Of this, Armour and Company have 17,000,000 cu. ft. in Chicago, and 9,500,000 cu. ft. in Kansas City, requiring 4380 and 3120 tons of refrigeration respectively; Morris and Company have 13,000,000 cu. ft. in Chicago and 8,000,000 cu. ft. in Kansas City, requiring 3450 and 2100 tons of refrigeration respectively.

Published statistics by the Bureau of Animal Husbandry of the United States Department of Agriculture, 1921, indicate that a total of 63,415,000 animals were slaughtered in 1920 under inspection, including 7,600,000 cattle, 3,800,000 calves, 13,000,000 sheep and lambs, and 39,000,000 hogs. At least 80 per cent of this total amount requires mechanical refrigeration for a short time at least, although the average time is between one and two months, and the average time after slaughtering before the meat is consumed is less than two months. A very small percentage (5 per cent) of the meat is frozen and is held in freezer storage.

The packing house practice has been a development, as in other applications of refrigeration. There are two main operations—the chilling of the hot freshly slaughtered animals and the maintaining of cold or freezer storage. In general "forced" draft or the forcing of chilled air through pipes or ducts is not used in American practice, because of the belief that it causes shrinkage, tends to darken the meat, that the increased pressure necessary to circulate the air causes excessive air leaks, and because of the expense for the operation of the fan and the space occupied by the air ducts. A modified system using "sprays" either in a loft, troughs, or in pipes is a modern successor to the old duct system.

The use of direct expansion or of brine is a matter for individual decision, although certain factors tend to swing the preference towards the use of brine. One of the important reasons is because of the danger in the use of direct expansion of ammonia by the class of labor to be found in the packing plant. Also the long pipe systems represent a heavy investment of ammonia—a large part of which would be lost (with a

## TABLE 98

| Suction pressure | | 45 | 40 | 35 | 30 | 27 | 25 | 22 | 19 | 15 | 12 | 10 | 7 | 4.7 | 1.4 | − 2.4 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Temperature of $NH_3$, deg. F. | | 30 | 26 | 22 | 17 | 14 | 12 | 8 | 4 | 0 | −5 | −8 | −13 | −18 | −25 | −35 |
| Temperature of brine, deg. F. | | 38 | 34 | 30 | 25 | 22 | 20 | 16 | 12 | 8 | 3 | 0 | − 5 | −10 | −19 | −30 |
| Room Temperature, Degrees F. | Piping | | | | | | | | | | | | | | | |
| 65 | 2 -in. Dr. Exp. | 220 | 198 | 179 | 160 | 151 | 145 | 135 | 126 | 119 | 110 | 106 | 99 | 93 | 86 | 77 |
| | 1¼-in. Dr. Exp. | 314 | 282 | 256 | 229 | 216 | 208 | 193 | 180 | 170 | 157 | 151 | 141 | 133 | 123 | 110 |
| | 1¼-in. Brine | 400 | 349 | 309 | 270 | 252 | 240 | 221 | 204 | 190 | 178 | 167 | 155 | 144 | 129 | 114 |
| 60 | 2 -in. D. E. | 257 | 227 | 203 | 179 | 168 | 161 | 148 | 138 | 129 | 119 | 114 | 106 | 99 | 91 | 81 |
| | 1¼-in. D. E. | 368 | 324 | 290 | 256 | 239 | 229 | 212 | 197 | 184 | 170 | 162 | 151 | 141 | 130 | 116 |
| | 1¼-in. Br. | 491 | 416 | 360 | 309 | 285 | 270 | 246 | 225 | 208 | 190 | 180 | 167 | 155 | 137 | 120 |
| 55 | 2 -in. D. E. | 308 | 266 | 234 | 203 | 188 | 179 | 164 | 151 | 140 | 129 | 122 | 113 | 106 | 96 | 86 |
| | 1¼-in. D. E. | 440 | 380 | 335 | 290 | 269 | 256 | 244 | 216 | 200 | 184 | 175 | 163 | 152 | 137 | 123 |
| | 1¼-in. Br. | 635 | 515 | 433 | 360 | 328 | 309 | 277 | 251 | 230 | 208 | 197 | 180 | 167 | 146 | 127 |
| 50 | 2 -in. D. E. | 365 | 321 | 275 | 234 | 214 | 203 | 184 | 168 | 154 | 140 | 133 | 122 | 114 | 103 | 91 |
| | 1¼-in. D. E. | 550 | 459 | 393 | 337 | 306 | 290 | 264 | 240 | 220 | 200 | 193 | 175 | 163 | 147 | 130 |
| | 1¼-in. Br. | 900 | 675 | 540 | 432 | 386 | 360 | 318 | 284 | 258 | 230 | 216 | 197 | 180 | 157 | 135 |
| 45 | 2 -in. D. E. | 514 | 406 | 335 | 275 | 249 | 234 | 208 | 188 | 171 | 154 | 145 | 133 | 123 | 110 | 96 |
| | 1¼-in. D. E. | 735 | 580 | 478 | 393 | 356 | 334 | 297 | 269 | 244 | 220 | 207 | 190 | 176 | 157 | 137 |
| | 1¼-in. Br. | 1540 | 982 | 720 | 540 | 470 | 432 | 373 | 327 | 292 | 258 | 240 | 216 | 197 | 169 | 144 |
| 40 | 2 -in. D. E. | 770 | 550 | 428 | 335 | 297 | 275 | 241 | 214 | 193 | 171 | 161 | 145 | 133 | 119 | 103 |
| | 1¼-in. D. E. | 1110 | 785 | 612 | 478 | 425 | 393 | 344 | 306 | 276 | 244 | 232 | 207 | 190 | 170 | 147 |
| | 1¼-in. Br. | 5400 | 1800 | 1080 | 780 | 600 | 540 | 450 | 386 | 338 | 292 | 270 | 240 | 216 | 184 | 154 |
| 35 | 2 -in. D. E. | 1540 | 856 | 592 | 427 | 366 | 334 | 286 | 248 | 220 | 192 | 179 | 160 | 145 | 129 | 110 |
| | 1¼-in. D. E. | 2220 | 1220 | 845 | 610 | 523 | 477 | 409 | 355 | 314 | 274 | 256 | 229 | 207 | 184 | 157 |
| | 1¼-in. Br. | ..... | ..... | 2160 | 1080 | 830 | 720 | 569 | 470 | 400 | 337 | 308 | 270 | 240 | 200 | 166 |
| 30 | 2 -in. D. E. | ..... | 1925 | 962 | 592 | 481 | 427 | 350 | 297 | 256 | 220 | 202 | 179 | 160 | 140 | 119 |
| | 1¼-in. D. E. | ..... | ..... | ..... | 845 | 688 | 610 | 500 | 424 | 366 | 314 | 288 | 256 | 229 | 200 | 170 |
| | 1¼-in. Br. | ..... | ..... | ..... | 2160 | 1350 | 1080 | 772 | 600 | 490 | 400 | 360 | 308 | 270 | 220 | 180 |
| 25 | 2- in. D. E. | ..... | ..... | 2566 | 962 | 700 | 592 | 452 | 367 | 308 | 256 | 233 | 202 | 179 | 154 | 129 |
| | 1¼-in. D.E. | ..... | ..... | ..... | 1375 | 1001 | 846 | 645 | 525 | 440 | 366 | 333 | 288 | 256 | 220 | 185 |
| | 1¼-in. Br. | ..... | ..... | ..... | ..... | ..... | 2160 | 1200 | 830 | 636 | 490 | 432 | 360 | 308 | 246 | 196 |
| 20 | 2 -in. D. E. | ..... | ..... | ..... | 2566 | 1283 | 962 | 642 | 482 | 385 | 308 | 275 | 233 | 202 | 171 | 140 |
| | 1¼-in. D. E. | ..... | ..... | ..... | ..... | ..... | 1375 | 917 | 689 | 550 | 440 | 393 | 333 | 289 | 244 | 200 |
| | 1¼-in. Br. | ..... | ..... | ..... | ..... | ..... | ..... | 2700 | 1350 | 900 | 636 | 540 | 432 | 360 | 277 | 216 |
| 15 | 2 -in. D. E. | ..... | ..... | ..... | ..... | 7700 | 2566 | 1100 | 700 | 514 | 385 | 334 | 275 | 233 | 193 | 154 |
| | 1¼-in. D. E. | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 1000 | 734 | 550 | 477 | 293 | 333 | 276 | 220 |
| | 1¼-in. Br. | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 3600 | 1543 | 900 | 520 | 540 | 432 | 318 | 240 |
| 0 | 2 -in. D. E. | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 7700 | 1540 | 962 | 592 | 527 | 308 | 220 |
| | 1¼-in. D. E. | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 1375 | 846 | 754 | 441 | 315 |
| | 1¼-in. Br. | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 2160 | 1080 | 569 | 360 |
| −10 | 2 -in. D. E. | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 2570 | 960 | 514 | 308 |
| | 1¼-in. D. E. | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 1375 | 735 | 441 |
| | 1¼-in. Br. | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | ..... | 1200 | 540 |

Factor "$K$" = B.t.u. transmitted per square foot of pipe per 1 deg. difference per hour.

Table is based on heat transmission of $K = 2.5$; if any other value of $K$ is desired multiply by 2.5 and divide by new value of $K$. If no frost accumulates on pipe, multiply lineal foot by 0.6. For forced air circulation multiply by .5 to .6.

forced shut-down) by accident to any part of the ammonia system, as well as likelihood of heavy replacement expense due to flange or other leaks. Brine is much more flexible, the temperature is easy to control, " off peaks " operation is possible, brine spraying is available and the piping is cheaper. When necessary, a − 20 deg. F. calcium chloride brine can be carried.

**The Balanced System.**—When brine is used the preferred method is to use a balanced system whenever possible. By this means the piping acts as a syphon and the pumping cost is that of overcoming the pipe friction only. By the proper use of the balance tank, the pipes are kept full of liquid all the time, and the pressure against which the pump has to work should be only 20 to 25 lb. for a nine story building. This is called a *closed* brine system. The cooler lofts sometimes have brine defrosting pipes, in which case some form of concentrator should be provided in order to conserve the salt. One method of removing the frost is to arrange a slotted 2-in. brine pipe over each bank, so piped that defrosting is possible by permitting brine to drop from pipe to pipe. Usually one-third of the coils are defrosted at a time. Heavy frost accumulations are very detrimental to heat transfer, and the refrigerating ability is thereby reduced. At times one hears that as the outside of the pipe becomes larger the total transfer of heat must remain practically constant. The same argument would be true in the case of pipe covered with magnesia or corkboard (as far as the outside diameter of the surface is concerned). The brine flowing over the pipes must be kept concentrated so as not to *freeze* on the pipes.

If an " open " brine system is used there is an advantage at times in installing the balance tank at a point best located to receive the drain from the brine spray nozzles. The closed pipe portion also empties into the tank and a booster pump supplies the extra pressure required for the sprays. Any open brine system will absorb moisture from the air and in the best chill rooms the shrinkage is at least $1\frac{1}{2}$ to 2 per cent. As the steam from the carcasses must be absorbed by the brine it is evident that the brine must be either thrown away or reconcentrated. One type of concentrator is a Baudalot arrangement where the brine passes *upward* inside 17 pipes, then up to a trough at the top of the stand and then downwards on the outside of six steam-heated pipes and then outside the 17 pipes already mentioned as a counterflow to the brine inside of the pipes. The six upper pipes provide the heating surface for the concentration, although it is evident that the next two or three are of some assistance in providing the time required for the removal of the steam. An open vat (steam heated) could be used, but if the exchanger principle is lost the design is not an economical one.

## TABLE 99

### Wholesale Meat Markets

*Refrigeration Data*

| | Horse-power | Tons | Cubic Feet | Kw. Hours per Year | Kw. Hours per Year per Cubic Foot | Cooler Temperature, Degrees F. |
|---|---|---|---|---|---|---|
| 1. Philadelphia, Pa. | 15 | 8 | 22,980 | 25,728 | 1.12 | |
| 2. Philadelphia, Pa. | 10 | 6 | 15,732 | 30,448 | 1.93 | |
| 3. Philadelphia, Pa. | 25 | 15 | 21,384 | 42,636 | 2.00 | |
| 4. Chicago, Ill. | 30 | 12½ | 23,190 | 30,456 | 1.31 | 20 |
| 5. Chicago, Ill. | 10 | 7 | 7,690 | 14,880 | 1.93 | 20 |
| 6. Pittsburgh, Pa. | 50 | 20 | 94,000 | 159,250 | 1.69 | 30 |
| *Packers* | | | | | | |
| 7. Chicago, Ill. | 30 | 15 | 4,340 | 6,000 | 1.38 | 36 |
| 8. Pittsburgh, Pa. | 30 | 15 | 39,680 | 76,292 | 1.93 | 36 |
| 9. Pittsburgh, Pa. | 30 | 15 | 37,450 | 171,144 | 4.61 | 40 (2 comp.) |
| 10. New Orleans, La. | 30 | 15 | 54,500 | 51,372 | 0.944 | 35 |
| | 5 | 2 | 1,440 | 7,580 | 5.27 | 40 |
| | 65 | 20 | 94,000 | 139,225 | 1.47 | 33 |
| *Provisions* | | | | | | |
| 11. New York, N. Y. | 15 | 8 | 10,505 | 23,412 | 2.24 | 36 |
| 12. New York, N. Y. | 10 | 4 | 38,880 | 25,783 | 0.665 | 38 (makes ice) |
| 13. Chicago, Ill. | 15 | 8 | 8,820 | 12,204 | 1.38 | 41 |
| *Retail Meat* | | | | | | |
| 14. Philadelphia, Pa. | 15 | 8 | 3,000 | 9,949 | 3.3 | 35 |
| 15. New Orleans, La. | 7 | 2 | 1,490 | 4,600 | 3.1 | |
| 16. Chicago, Ill. | 15 | 8 | 4,629 | 26,808 | 5.8 | 36 |
| 17. Chicago, Ill. | 5 | 2 | 1,706 | 3,408 | 2.0 | 43 |
| 18. Philadelphia, Pa. | 7½ | 4 | 2,500 | 11,099 | 4.45 | |
| *Fur Storage* | | | | | | |
| 19. Brooklyn, N. Y. | 39¼ | 20 | 56,775 | 19,760 | 0.35 | 30 |
| 20. New York, N. Y. | 7½ | 4 | 5,722 | 1,931 | 0.34 | 20 |
| 21. New York, N. Y. | 15 | 10 | 33,120 | 23,000 | 0.70 | 16 |
| 22. New York, N. Y. | 30 | 15 | 40,500 | 27,321 | 0.67 | 32 |

**Chilling.**—The best practice at present is to place the beef, sheep, and veal immediately in the chill rooms after the hides have been removed. The maximum time interval for standing in the air is a half hour. This chill room or fore-cooler is at 45 to 48 deg. F., and an attempt is made to bring the temperature down to 38 degrees in 12 hours. The feeling prevails in Chicago that hogs can be air dried over night and are improved in the process but careless handling may cause sour meat. The use of fore-coolers to separate freshly killed carcasses from chilled ones is still a disputed practice, but it is favored by many if space will permit it. The meat should hang 12 in. apart in the chill rooms, but heavy cattle should be 18 in. apart in order to provide circulation. The fore-cooler, if used, should be kept as near 38 deg. F. as possible, although 43 degrees is not bad for beef. If the main cooler is fitted for a fore-cooler it should be held at a maximum temperature of 38 deg. F. and it should be cooled to 34 deg. F. in 12 to 15 hours, and this 34 degrees should be held. Heavy cattle and frozen meat for export have special treatment. For example, heavy cattle should be carried at 32 deg. F. for 24 hours, then at 34 deg. F. if used for domestic trade. For export, reduce to 29 degrees as quickly as possible, and keep at that temperature until "floated." Mutton and veal are handled about the same as beef. Pork is air cooled (for the first two hours at least), and is hung in a 32-degree room which is not allowed to rise above 38 degrees. This room temperature is reduced to 32 degrees in 12 hours.

**Refrigeration.** The calculation of the refrigeration load is usually by rule of thumb. The daily kill[16] is a favorite method, as one ton of refrigeration is required in the ratio of the following number of animals slaughtered daily:

5 to 6 cattle at 700 lb.;
40 to 50 calves at 80 lb.;
15 to 22 hogs at 225 lb.;
50 to 65 sheep at 60 lb.;
1000 to 1300 chickens at 3 lb.

This rule takes for granted that the rooms are properly insulated. Where the piping is 2-in. direct expansion the packing house rule is:

| | | | | | | |
|---|---|---|---|---|---|---|
| Warm beef cooler | requires 1 ft. of 2-in. pipe per | | | 10 | cu. ft. | of storage. |
| Beef cooler | " | " | " | 12 | " | " |
| Sheep cooler | " | " | " | 12 | " | " |
| Hog cooler | " | " | " | 8 | " | " |
| Sausage (at 30 deg. F.) | " | " | " | 10 | " | " |
| Curing cellars (36 deg. F.) | " | " | " | 18 | " | " |
| Freezers (15 deg. F.) | " | " | " | 6 | " | " |
| Freezers (0 deg. F.) | " | " | " | $2\frac{1}{2}$ | " | " |

[16] Accepted practice, authority unknown.

A rough rule requires one ton of refrigeration per 7000 to 12,000 cu. ft. of space at 33 to 38 deg. F. As an illustration of the working of these rule-of-thumb rules, the following may be taken as an example:

Find the refrigeration required for 100 cattle daily killed, and hanging space for 400 cattle, 6500 sq. ft. of cooler space required with 33 ft. ceiling or 143,000 cu. ft. of space; for 250 hogs daily or 750 hanging space allowing 3000 sq. ft. at 18 ft. ceiling or 54,000 cu. ft. and a storage for a total turnover in 60 days (storage for 1,500,000 lb. of meat product)

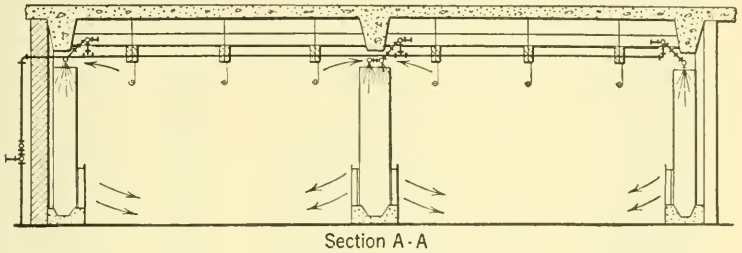



Fig. 294.—Air Cooling with Brine Sprays.

requiring 15,600 to 20,000 sq. ft. of floor and 200,000 cu. ft. of space. As a by-product there will be 10,000 lb. lard daily.

| | Tons |
|---|---|
| 143,000 cu. ft. of beef cooler at 1 ton per 10,000 cu. ft. | 14.3 |
| 100 cattle daily at 500 lb. | 12.2 |
| 54,000 cu. ft. hog cooler space at 1 ton per 10,000 cu. ft. | 5.4 |
| 250 hogs at 180 lb. | 10.9 |
| 200,000 cu. ft. cellar space at 1 ton per 12,000 cu. ft. | 16.6 |
| 15 tons of ice at 2.0 tons of refrigeration | 30.0 |
| 10,000 lb. of lard (4 hours' duration) | 12.0 |
| 100,000 cu. ft. freezer space at 1 ton per 3000 cu ft. | 35.0 |
| Total | 136.4 |

In all likelihood rough rules like the preceding have been satisfactory inasmuch as the refrigeration supplied would be much greater than the actual requirements in order to permit the installation of sufficient spares to insure continuity of operation in the time of breakdown. It seems likely, however, that in the future the refrigeration requirements will be more carefully calculated, i.e., a determination of the heat leakage and the live load.

Brine sprays have been used in the packing plant for years with varying success. The older designs used a semi-solid spray which provided circulation but was not efficient. During the chilling process a large amount of steam is thrown off into the air, the shrinkage of weight

being usually 1½ to 2 per cent by weight of the carcass, but sometimes as much as 3 or 4 per cent. The packer desires to reduce this loss to a minimum because of the loss of weight of the product, as well as the difficulty met with at times from the large amount of frost accumulation on the pipes. The chill room is always made with a loft in order to get a lively circulation of air, and circulation is always required if quick cooling is in demand. Such lofts are shown in Figs. 292 and 299, giving an example of the arrangement of piping and sprays.

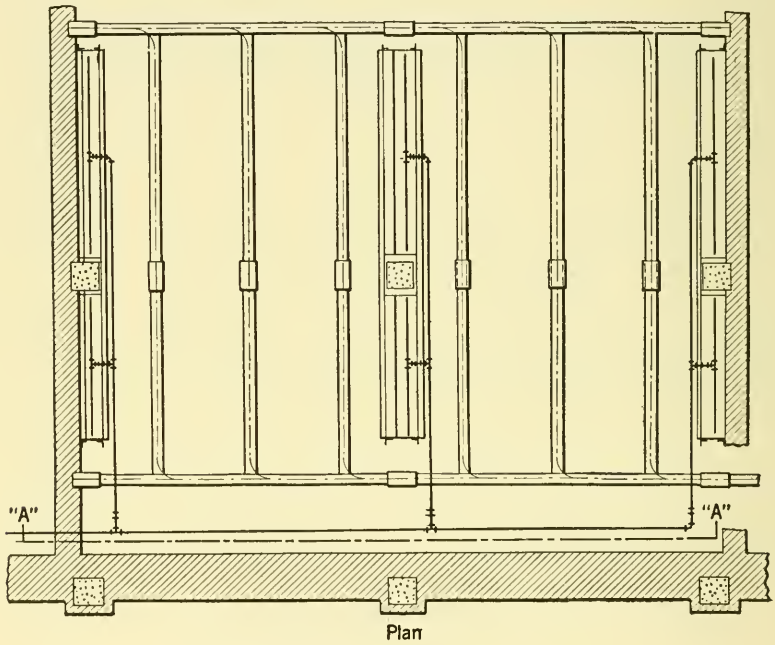



Fig. 295.—Air Cooling with Brine Sprays.

**The Spray System.**—The best results with sprays[16a] are obtained with atomizing nozzles. With such sprays, hogs dressed at 225 lb. can be chilled to inside ham temperatures of 36 deg. F. in 24 hours with 25 deg. brine, using 30 lb. suction pressure at the compressor. Beef can be chilled to from 34 to 36 deg. F. (inside) in 36 hr. using weak brine or possibly *water sprays* at 33 degrees and at a great saving in the shrinkage. The great advantage in the use of sprays is the quick removal of the moisture in the air, the rapid circulation and the ready capacity variation by decreasing the nozzle pressure, or the brine or the water temperature.

[16a] Samuel Bloom, The National Provisioner, February, 1924. American Society of Refrigerating Engineers, Journal, 1921.

The objection is the shrinkage and the danger of the mists carrying over and being deposited on the meat. Up to the present, water in the sprays of chill rooms has not been found practical, and the result is that *salt* brine (NaCl) is always used. In order to decrease the shrinkage the humidity is kept as high as possible, requiring a low salt concentration and therefore a relatively high temperature of the brine.

Mr. Bloom advocates that the height of the loft be made proportional to the load, the width of the deck and the temperature of the brine, but he considers that it should never be over 6 ft. nor less than 2 ft. (because of insulation difficulties and waterproofing of the deck). The width of the two openings in the deck should be equal to the loft height, and the deck should be insulated with 2 in. of corkboard which should be waterproofed very carefully.

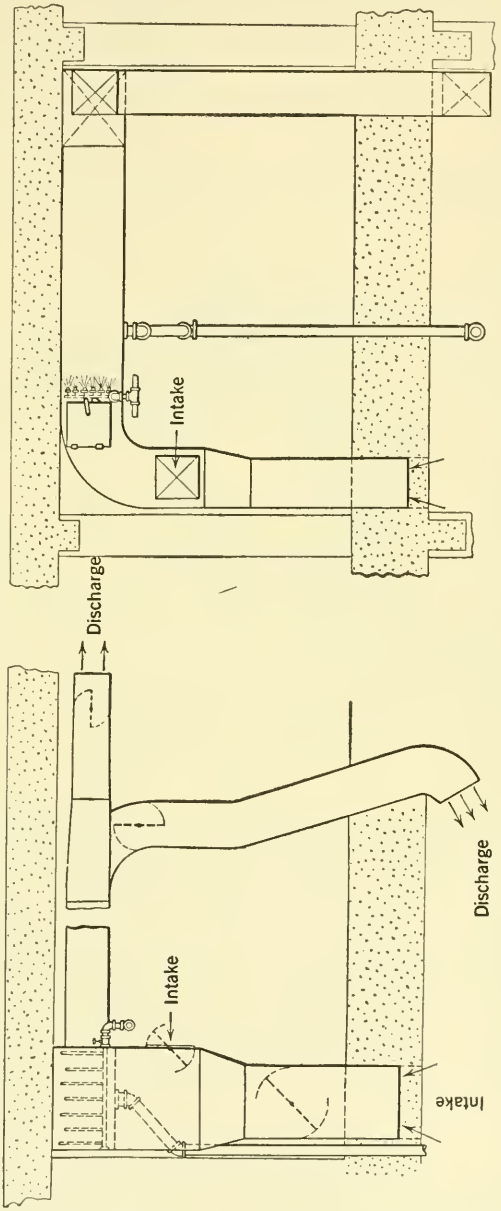



FIG. 296.—Air Cooling with Brine Sprays.

The difficulty with dirt in the brine is overcome by the operation of leaching the salt on solution, and by having settling tanks which may be cleaned out every year. The

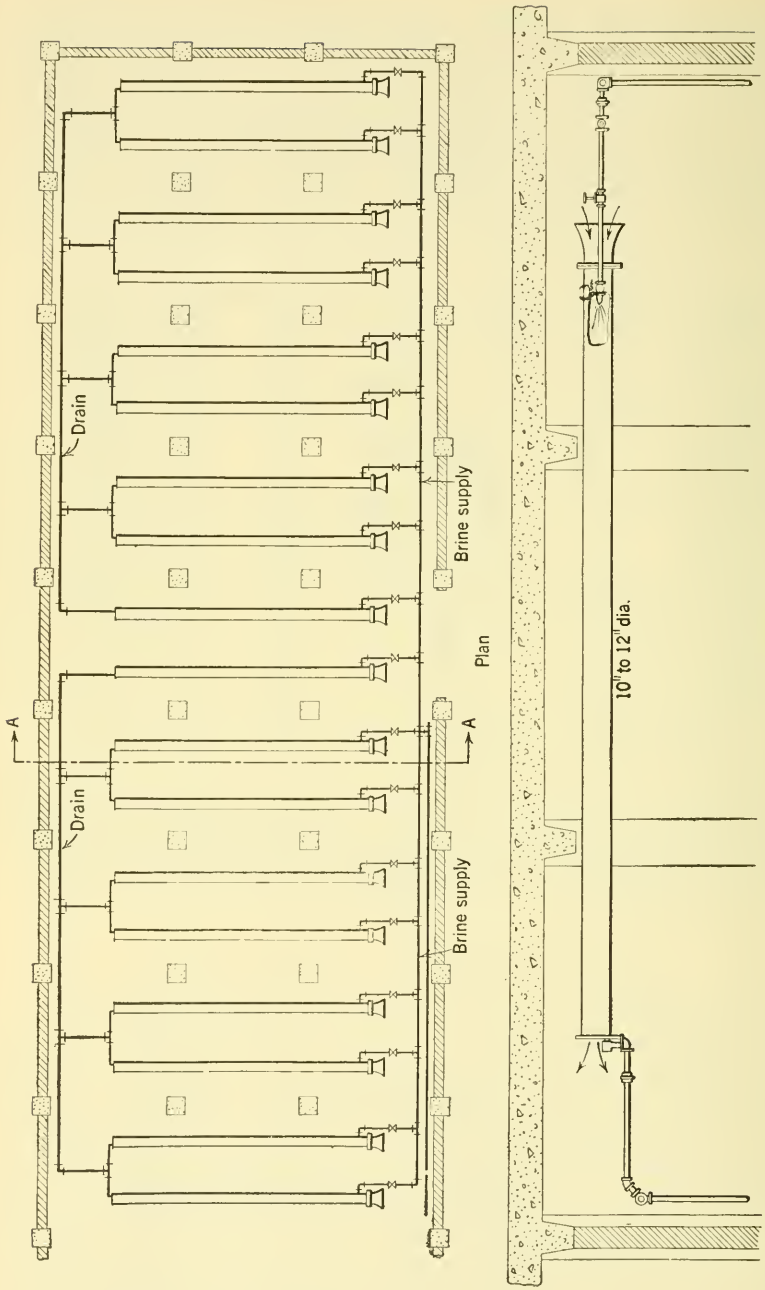



FIG. 297.—Air Cooling with Brine Sprays.

brine pressure at the nozzle should be from 9 to 2 lb. per sq. in. minimum and the decks should be from 38 ft. to 9 ft. respectively. Too narrow a deck will permit the droplets of brine to carry over, for even if they are in the air only 3 seconds at a velocity of 600 ft. per min. the distance carried will be 30 ft. The height of the spray above the deck and the velocity of the air are the determining factors as regards the width of the deck.

To cool the brine the shell and tube brine cooler is the preferred means because of its great flexibility, high heat transfer and relative lack of resistance to brine flow if the number of passes are small. The shell and tube as well as the double pipe brine cooler must be watched carefully to prevent injury due to freezing of the brine should the pump stop. These coolers when placed on the *suction* side of the pump can be made safe provided a means of draining the cooler is arranged for in the event of the stopping of the pump. Calcium chloride brine is never used for sprays in the packing house because of the drying action of the calcium brine, because of the greater cost and of the danger of the mists carrying over and being deposited on the meat, thus destroying palatability. The best arrangement is to provide a float-operated tank in order to keep a constant head on the nozzles. If a centrifugal pump is used it should be such a one as will give a "flat" characteristic curve—enabling one to get the same pressure on the nozzle for a range of capacity of the pump (see Chapter VIII).

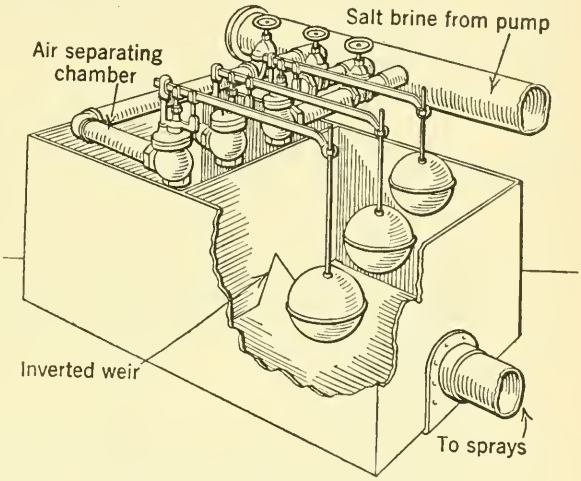
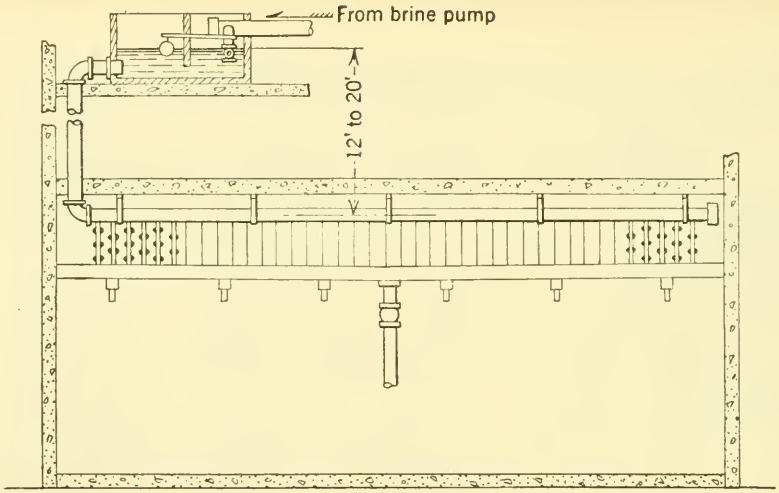
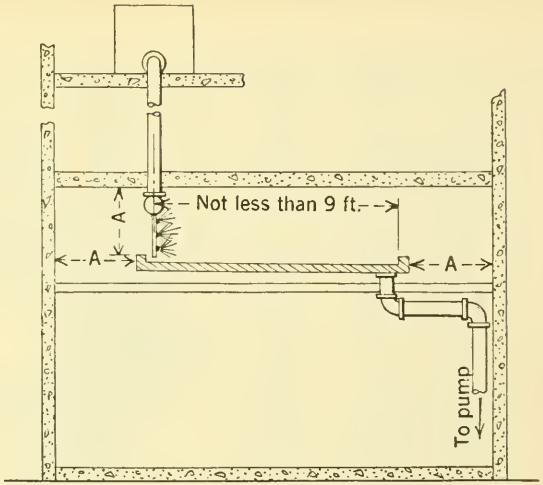



FIG. 298.—Gravity Brine Tank.

**Problem.**—As an example when an attempt at a close calculation is to be made, let it be required to chill 50 beeves at 750 lb. each from a temperature of 90 deg. to 38 deg. F. in 12 hours. The brine in the fore-cooler will enter the spray at 32 degrees




| Press. at Nozzle lb/sq. in. | Gallons per Minute | | |
|---|---|---|---|
| | Webster 3/16" open | Braemer #30 | Carrier 3/16" open |
| 6 | 1.09 | | 0.88 |
| 9 | 1.25 | | 1.10 |
| 12 | 1.41 | 1.55 | 1.33 |
| 21 | 1.87 | 2.05 | 1.84 |
| 24 | 2.00 | 2.19 | 1.98 |
| 27 | 2.12 | 2.32 | 2.09 |
| 30 | 2.24 | 2.45 | 2.20 |
| 33 | 2.34 | 2.57 | 2.32 |
| 36 | 2.45 | 2.60 | 2.40 |
| 39 | 2.55 | 2.80 | 2.50 |

FIG. 299.—The Gravity Brine Tank.

and will rise in temperature to 35 degrees. The regular spray deck will be used, and salt brine only at as low a concentration as possible will be made use of.

| | | |
|---|---|---|
| To cool the meat | $50 \times 750 \times 0.77 \times (90 - 38)$ = | 1,501,500 B.t.u. |
| | = | 10.43 tons |
| Allowing 20 per cent for losses (these could be calculated if the details of the insulation, etc., were known | = | 2.09 tons |
| Total | = | 12.52 tons |

Allowing 12 sq. ft. per animal there will be required $12 \times 50 = 600$ sq. ft. floor and a room 20 ft. by 30 ft. will answer the conditions. The usual distance between rails is 4 ft. 0 in., and the distance from the rail to the floor 11 ft. 2 in. (Fig. 299). The amount of brine required will be as follows:

$$\text{Wt.} \times 0.892 \times 3 = 12.52 \times 200 \quad = 2504 \text{ B.t.u. (per minute)}$$
$$\text{Wt. of brine per minute} = 2504 \div 2.676 = 936 \text{ lb.}$$

Using a specific gravity of the brine of 1.066, the gallons of brine required will be $936 \div (8.33 \times 1.066) = 105.4$. Under a head of 13.05 ft. (6.0 lb.) the No. 30 Webster nozzle will deliver 1 gal. per min. and there will therefore be required 105 nozzles. The maximum spacing of the nozzles is 20 in., so the placing of these four high on 10-in. centers, and 26 stands equally spaced horizontally (less than 14 in. apart), will be a good distribution. The height of the deck will be 3 ft. 6 in., and this will also be the size of the two openings in the deck. This will provide a vigorous circulation of the air of about two changes per minute, there being no choking at any point.

**Lard and Compound Cooling.**—Lard weighs about 59.3 lb. per cu. ft., the specific heat is from 0.5 to 0.6 depending on the fluidity, and the latent heat is 90 B.t.u. per lb. The melting point is usually understood to be 70 deg. F. for lard and 105 deg. for tallow.

Lard compound, a mixture of hog fat with beef fat and cottonseed oil, has a specific heat of from 0.3 to 0.5 depending on its fluidity, a latent heat of 90 B.t.u. per lb. and a specific gravity of 0.92 (57.5 lb. per cu. ft.).

Mechanical refrigeration is used in the cooling of both lard compound and lard. The manner of applying refrigeration is to use hollow cylindrical drums, which are maintained at a temperature of about 10 deg. F. by means of brine or direct expansion, and which are made to revolve at from 10 to 14 r.p.m. The usual custom is for the lard or the compound to be cooled first with water from 135 to 140 deg. F. to 80 degrees and then by passing the commodity to the cooling rolls to cool it to the final temperature of from 30 to 40 degrees.

In the manufacture of lard the object of refrigeration (besides the speed of the operation which is very important) is the uniformity and better quality of the product. In the manufacture of lard the trimmings

and other fat parts of the hog are steamed in a rendering tank under 40 lb. steam pressure for some 7 to 9 hours. It is then mixed with fuller's earth for a few minutes, and next filtered by passing it through a filter press. The resulting hot, white lard is stiffened by adding not more than 5 per cent of lard stearine, and then by cooling mechanically.

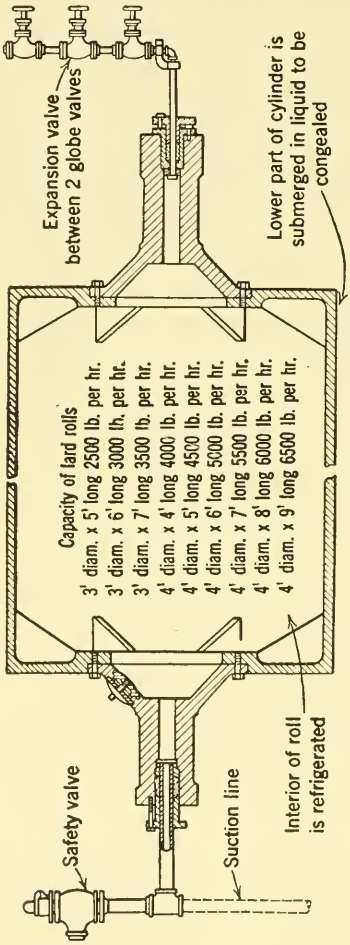

Expansion valve between 2 globe valves
Lower part of cylinder is submerged in liquid to be congealed
Capacity of lard rolls
3' diam. x 5' long 2500 lb. per hr.
3' diam. x 6' long 3000 lb. per hr.
3' diam. x 7' long 3500 lb. per hr.
4' diam. x 4' long 4000 lb. per hr.
4' diam. x 5' long 4500 lb. per hr.
4' diam. x 6' long 5000 lb. per hr.
4' diam. x 7' long 5500 lb. per hr.
4' diam. x 8' long 6000 lb. per hr.
4' diam. x 9' long 6500 lb. per hr.
Interior of roll is refrigerated
Suction line
Safety valve

Fig. 300.—Section of a Lard Roll.

Compounds are made using beef fat, cottonseed oil and pork fat. In making such a compound the refined fats are mixed and filtered, and then sent to the cold revolving drum. The drum chills the warm fluid quickly, thereby giving a uniform compound. Should slow cooling be permitted the beef crystals would freeze out first and non-uniformity would result. The revolving drum, cooled with brine or ammonia, freezes a thin layer of lard or compound, as the case may be, and, after the commodity has been on the surface for about three-quarters of a revolution of the drum, a carefully adjusted knife scrapes it off the surface. The semi-solid substance drops into the picker trough where thorough mixing is again brought about (Fig. 301). The product is now ready for the packages. Figure 300 shows the lard cooling cylinder in section, and Fig. 302 shows the entire process. As an example of the refrigeration necessary let it be required to find the amount of cooling effect necessary to handle 10,000 lb. of lard in four hours with brine. Also let it be required to find the size of the compressor and the size of the required cooling cylinder.

**Problem.**

$$10{,}000 \times 0.6 \times (80 - 30) + 10{,}000 \times 90 = 1{,}200{,}000 \text{ B.t.u. per 4 hours,}$$
$$= 300{,}000 \text{ B.t.u. per hour.}$$
$$300{,}000 \div 12{,}000 = 25 \text{ tons of refrigeration.}$$

Adding 10 per cent for losses = 27.5 tons of refrigeration required. As the compressor will be required to cool brine to + 10 deg. F., a zero degree boiling temperature of the ammonia will be required, and this corresponds practically to 15 lb. gage pressure suction to the compressor. With such a suction pressure there would be required 4.63 cu. ft. piston displacement per ton of refrigeration per minute using a volumetric efficiency of 0.83 and the piston displacement for 27.5 tons

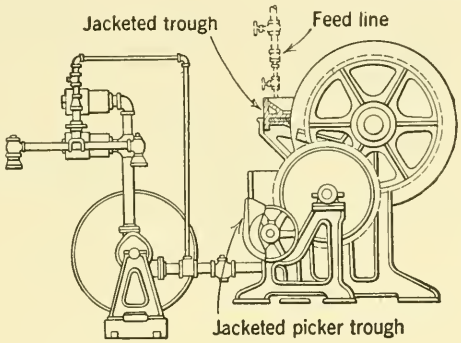



FIG. 301.—Assembly of Lard Rolls.

would be 128 cu. ft. per min. Referring to Fig. 300, which gives the construction of cylindrical drums, it will be seen that the size 3 ft. diam. and 5 ft. long will handle 2500 lb. per hour. This will be the required size. The thickness of the lard on the surface of the roll under these circumstances is less than 1/50 in.

**The Refrigerator Car.**—It has been found in cold storage practice that *only* the best construction is the cheapest, and that a rough rule of

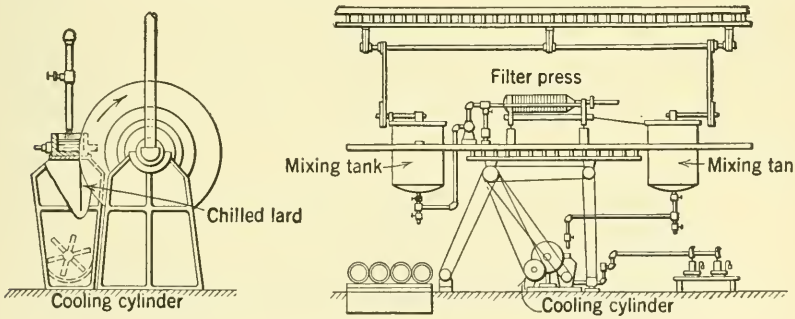



FIG. 302.—Lard Manufacture.

1 in. of corkboard or its equivalent is required for every 10 degrees of depressed temperature, and the same rule (slightly modified) applies also to the refrigerator car. For an average of 70 degrees outside and 40 degrees inside there would be required, according to this rule, 3 in. of corkboard. W. H. Winterrowd [17] states that calculated values for heat

[17] Some Notes on Railway Refrigerator Cars, Mechanical Engineering, July, 1922.

transfer for typical refrigerator constructions vary from 1.7 to 2.33 on the roof, 2.17 to 2.89 on the wall, and 2.46 to 2.54 on the floor, B.t.u. per sq. ft. per 24 hours. The value for 3-in. corkboard alone is about 2.1 without allowing for air spaces and the building construction. The

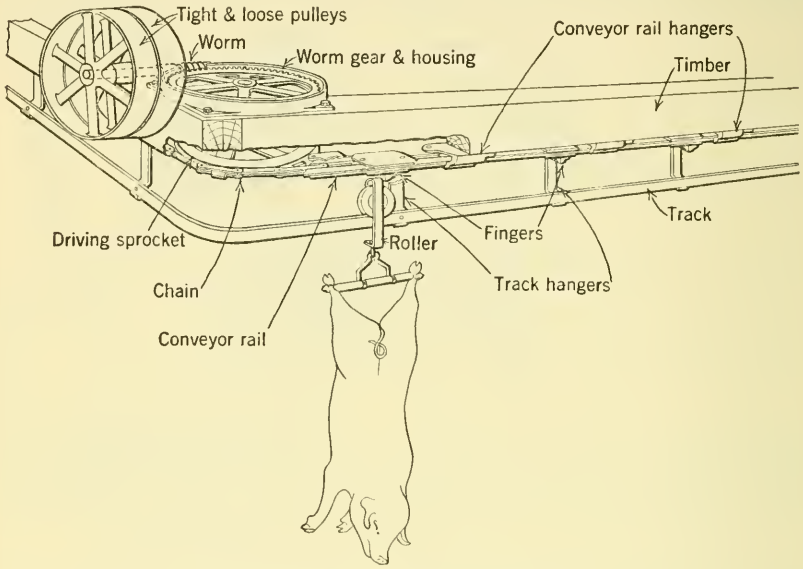



Fig. 303.—Overhead Conveyor.

refrigerator car and cold storage buildings are not comparable, however, as the car has a very rough service and the shock and vibration is such as to make it difficult to keep the car tight. The design of the car then must be one that will resist heat transfer, and will have a long life of service.

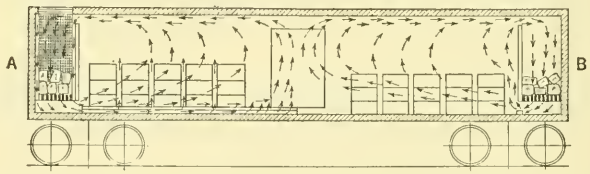



Fig. 304.—Air Circulation in a Refrigerator Car.

Next to heat transfer is the method of cooling and the efficiency of the air circulation (Fig. 304). The lading is continually developing heat which must be absorbed or cause a rise in the temperature. Cooling is applied in two principal ways: the best known and the most usual

method is the ice bunker at the two ends of the car, and the later design embodies a brine tank built into the roof at each end of the car, the two tanks being connected by pipes hung some two or three inches beneath the ceiling and so arranged that check valves in the tanks which are operated by the swaying of the car causes a positive flow of the brine. Ice and salt, of course, are charged into the tanks. The advantages claimed are more economy of space, and more even car temperatures. It is not clear what efficiency is obtained during the periods when the car is still.

TABLE 100

[By the U. S. Bureau of Agriculture Economics]

CARLOAD SHIPMENTS OF DOMESTIC FRUITS AND VEGETABLES

| Commodities | Number of Carloads | | | | | | |
|---|---|---|---|---|---|---|---|
| | 1917 | 1918 | 1919 | 1920 | 1921 | 1922 | 1923 |
| Apples | 57,048 | 68,840 | 81,552 | 102,962 | 96,498 | 101,780 | 127,505 |
| Asparagus | 1,057 | 1,120 | 1,101 | 1,034 | ..... | 746 | |
| Cabbage | 20,354 | 28,661 | 24,982 | 31,020 | 31,718 | 40,065 | 36,395 |
| Cantaloups | 17,430 | 13,619 | 22,039 | 22,377 | 25,569 | 29,917 | 25,839 |
| Carrots | ..... | ..... | ..... | ..... | ..... | 1,987 | |
| Cauliflower | ..... | 1,535 | 2,546 | 3,410 | 12,483 | 14,151 | 16,624 |
| Celery | 6,577 | 7,412 | 6,449 | 9,308 | 3,895 | 3,991 | 4,714 |
| Cherries | 897 | 680 | 1,025 | 1,870 | ..... | 2,447 | |
| Cranberries | ..... | ..... | ..... | ..... | ..... | 2,009 | |
| Cucumbers | 3,711 | 2,757 | 4,306 | ..... | ..... | 6,151 | |
| Eggplant | ..... | 300 | 136 | ..... | ..... | 271 | |
| Grapes | 21,379 | 20,915 | 30,349 | 39,205 | 37,202 | 59,858 | 64,020 |
| Lettuce | 5,428 | 6,959 | 8,018 | 13,818 | 18,616 | 22,840 | 29,493 |
| Onions | 19,152 | 22,027 | 20,874 | 25,950 | 23,319 | 27,563 | 26,747 |
| Peaches | 27,237 | 20,409 | 30,923 | 26,967 | 27,300 | 38,291 | 33,630 |
| Pears | 11,468 | 10,311 | 10,158 | 14,950 | 12,821 | 20,138 | 18,144 |
| Green peas | ..... | 691 | 278* | | | | |
| Peppers | ..... | 1,170 | 913 | ..... | ..... | 2,322 | |
| Plums and prunes | ..... | 3,125 | 3,222 | 4,430 | ..... | 5,021 | 6,786 |
| Sweet potatoes | 8,898 | 9,254 | 13,725 | 16,254 | 19,071 | 20,576 | 19,793 |
| White potatoes | 144,656 | 169,462 | 181,277 | 179,149 | 219,426 | 245,221 | 238,983 |
| Spinach | 1,278 | 2,422 | 2,957 | 2,687 | 4,924 | 5,205 | 7,682 |
| Strawberries | 15,065 | 8,452 | 8,105 | 8,490 | 10,695 | 18,716 | 17,775 |
| Tomatoes | 14,115 | 15,471 | 14,503 | 15,556 | 17,199 | 26,668 | 24,002 |
| Turnips | ..... | ..... | ..... | ..... | ..... | 835 | |
| Mixed vegetables | ..... | 8,804 | 11,933 | 12,740 | 15,816 | 19,676 | 24,016 |
| Watermelons | 31,503 | 20,392 | 30,860 | 39,255 | 46,463 | 47,066 | 35,405 |
| Deciduous fruits: | | | | | | | |
| Mixed | ..... | ..... | ..... | ..... | ..... | 8,609 | 8,998 |
| Citrus fruits: | | | | | | | |
| Grapefruit | 393 | 5,650 | 6,624 | 12,086 | 12,275 | 14,182 | 19,150 |
| Lemons | 401 | 6,913 | 8,823 | 9,718 | 11,887 | 9,875 | 8,668 |
| Oranges | 5,437 | 28,444 | 49,321 | 53,041 | 65,891 | 46,526 | 74,948 |
| Miscellaneous | ..... | 3,351 | 3,024 | 3,902 | 5,716 | 2,088 | |
| | 413,481 | 489,236 | 580,026 | 650,179 | 718,784 | 844,191 | 866,317 |

* Incomplete.

The following paper on "Railway Car Icing Systems" was presented by A. L. Blatti before the annual meeting of the National Association of Practical Refrigerating Engineers, held in Memphis, December 12 to 15, 1923.

It has been found that a platform longer than 1000 ft. is not practicable in some cases, because the time consumed in moving ice on such a platform more than offsets the advantage of having more cars at the dock than a 1000-ft. platform will accommodate. A platform of 1000 ft. length will accommodate 46 cars, or 23 cars on each side, and when more cars than this number arrive in a single train, it is customary, after splitting the train into two groups of cars which are placed on each side of the dock, to ice the group having the surplus cars first, and then while icing the other group of cars on the opposite side of the dock, the switching crew moves into place the hang-over or surplus cars, which are in turn iced without cessation of the entire operation or delay in the work of the icing crew.

Our experience has been that if the cakes of ice employed are of 300-lb. size, the icing platform should be 14 ft. wide, while if a 400-lb. cake is used it should be 16 or 17 ft. wide. The platform should be 14 ft. 6 in. above the top of the car rail, so that there will be sufficient fall from the platform to reach the far side of the bunker opening by means of a skid which will clear the running board on top of the car. The platform should, of course, have a conveyor chain running its entire length through its center and operated by reversible motor, so that ice may be moved backward and forward at will to any selected place on the platform.

The platform described is for use in placing ice in the bunkers of cars which contain fruit or vegetables and, therefore, do not require temperatures approximating the freezing point, so that cake ice may be employed. The icing of cars which contain packing-house products demanding a lower temperature, requires a different process, as crushed ice mixed with salt must be used for this purpose, the quantity of salt ranging from 10 to 15 per cent by weight of the crushed ice. For meat car icing a second deck to the platform previously described may be constructed, 7 ft. above the top of such platform. This upper deck will have no conveyor chain, but will provide a run-way for carts holding from 900 to 1000 lb. of crushed ice each, which carts may be moved upon the upper deck to the position required and the crushed ice contained in those carts dumped into movable traveling chutes, which are attached to the edge of the upper deck and which chutes are moved into place so that the lower end of the same point directly into the bunkers of meat cars being iced. Salt bins should be provided at intervals on the lower deck, so that they are readily available to the icing crew who may shovel such salt as is required into the bunker of the meat cars, after the crushed ice has been dumped into the bunker through the chute from the upper deck.

The average time required to ice vegetable or fruit train with cake ice varies from 1 to $2\frac{1}{2}$ minutes for each car, according to the amount of ice required in each bunker, while for icing of meat cars with crushed ice $2\frac{1}{2}$ minutes per car is a good average time because of the fact that additional time is required in draining the ice compartment of a meat car and working down the ice which may remain in the same, preparatory to the reception of additional crushed ice.

Considerable attention has been given to details in car construction during the last ten years and particularly since 1917. Figure 304 shows in "*A*" the basket type and in "*B*" the regular bunker type of box

bunker. The latter is now obsolete on account of the obstruction to the free circulation of air, poor cooling of the air and the resulting higher air temperatures in the car. The *basket* type is made of wire mesh with a slatted wooden structure, so arranged as to permit air to flow all around the basket. Referring to Fig. 304, it will be seen that the left-hand side has a false floor. This rack is arranged so as to give some 3 in. to 4 in. of space for the flow of air and is arranged with hinges at the sides of the car and is divided along the middle of the car so that it can be swung up against the wall. The probable air circulation is shown

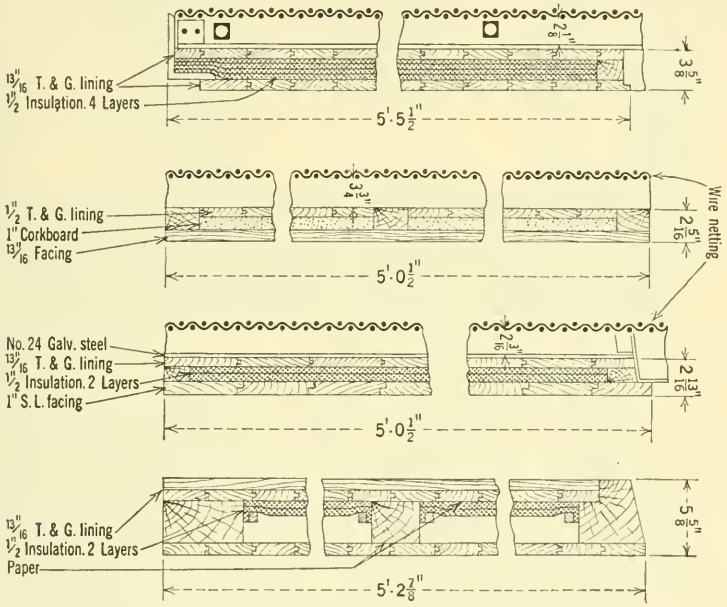



Fig. 305.—Insulation of Refrigerator Cars.

for the types, the slatted design giving the better results as would be expected since air can travel to the middle of the car whereas it becomes short-circuited with the construction shown on the "*B*" side of the car. The center of the car is the most severe test of refrigeration, and the length of the car is an important factor in the refrigeration efficiency. The general tendency is to have a length between bulkheads of from 32 to 34 ft., some 6 to 10 in. from the rack to the bottom of the opening in the bulkhead, and from 12 to 18 in. from the top of the bulkhead to the ceiling. The construction of the bulkhead is important, for otherwise freezing of the lading might occur near it. Figure 305 gives an idea of the modern trend.

The construction of the walls, roof and floor is given by Figs. 304 to 307. In the paper by W. H. Winterrowd it was brought out that corkboard is used in but few cases, except on the floor where 2-in. corkboard is usual. The feeling seems to prevail that corkboard does not endure

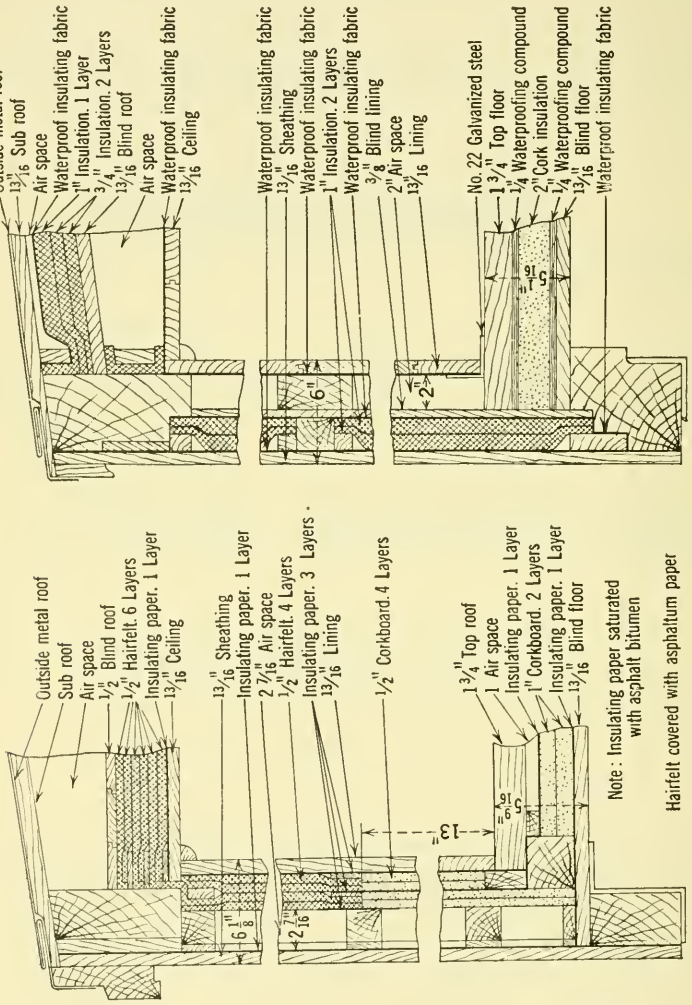



FIG. 306 and 307.—Insulation of Refrigerator Cars.

as well as more flexible insulation, but that the insulation must be suitably protected from moisture. The floors and the walls should be constructed so as to permit the driving of nails without injury to the insulation.

## REFRIGERATOR CARS IN OPERATION

Information Bulletin, No. 345, of the Perishable Freight Conservation Bureau, American Association of Ice and Refrigeration, quotes some figures from railway equipment register concerning the number of refrigerator cars in operation. The figures are as follows:

| | |
|---|---|
| U. S. Railroads | 32,773 |
| Car Lines (Railroad controlled) | 94,007 |
| Car Lines (Independent) | 5,855 |
| Meat Packers | 18,650 |
| Canadian Railroads | 6,093 |
| Mexican | 2 |
| Total | 157,380 |

Installed in Service:

| | | First 9 mos. |
|---|---|---|
| | 1923 | 1924 |
| Railroads | 7,570 | 3,801 |
| Private | 14,508 | 7,875 |
| Total | 22,078 | 11,678 |

According to the figures for the first nine months of 1924, there will be quite an appreciable decrease in the number of cars installed in service for the year 1924.

**Mechanical Refrigeration.**—Many attempts have been made to refrigerate freight cars mechanically. However, the difficulties are so many that it seems improbable that success will be obtained for American cars for some time. A mechanical drive from the axle seems to be impossible, considering the frequent delays. The central plant car to be designed for ten or more cars could be operated by a standard oil engine, the engine and compressor to be air cooled, probably using a water cooled condenser and an automobile type of air cooled radiator with improved fan ventilation for cooling the water. This would enable the water to be conserved and at the same time would permit better heat transfer in the condenser..

**Precooling of Fruits and Vegetables.**—Most fruits and some vegetables are shipped by means of refrigerator cars from three to six days' carry. If the fruit is warm (say 80 to 90 deg. F.) when loading into the freight car it takes from 3 to 5 days[18] or more to get the temperature of the inner packages down to a temperature of 40 to 45 deg. F. and in many cases this degree of temperature is never reached, while in any case the temperature is very uneven throughout the car. Much fruit is

[18] Bulletin No. 123, Bureau of Plant Industry, Department of Agriculture.

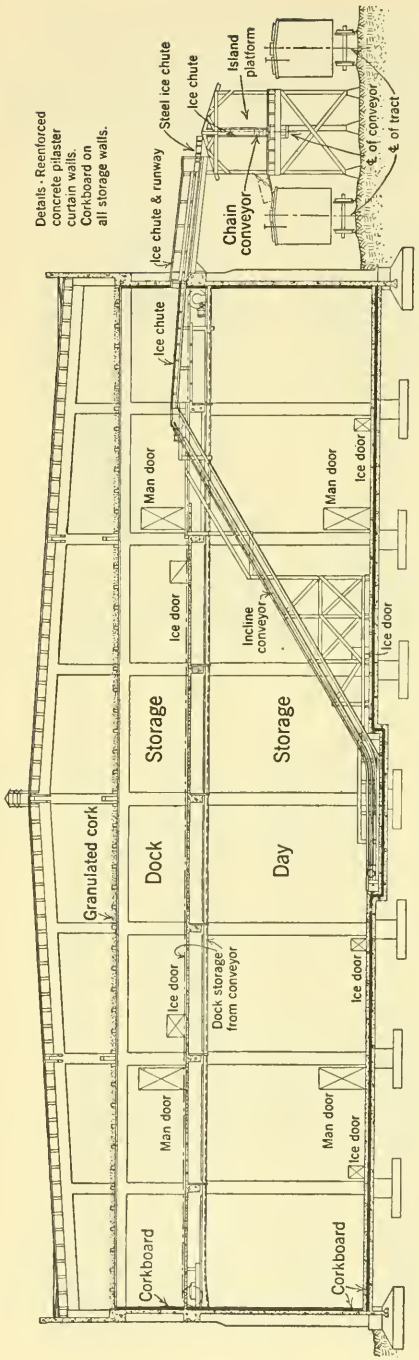



Fig. 308.—Refrigerator Car Icing.

spoiled, or the upper layers of packages have a secondary quality as compared with those on the floor near the ends of the car. Not only does the fruit spoil by the action of the fungi, but the resulting formation of carbon dioxide tends to increase the temperature. For example, it is stated that peaches at 86 deg. F. will rise 18 degrees in 20 hours, while at 68 degrees the increase is only 6 degrees in the same interval of time. A rough calculation gives an ice meltage of 22 per cent due to the self-heating of the fruit. It is evident then that shipment of perishable fruits involves the following important phases:

*a.* The development of a high grade refrigerator car.

*b.* Means of properly precooling perishable fruits and vegetables.

*c.* Economical means of car icing at convenient points of train departure and suitable re-icing depots at division points along the road.

The importance of proper precooling or icing or both can be well brought out by the history of the expansion and development of the Pacific Fruit Express Company as an example of what the industry has done in that respect in the last 20 years.

In 1907 the Pacific Fruit Express Company (operated jointly by the Union Pacific and the Southern Pacific Railroads) owned 6600 refrigerating cars, handling 48,900 carloads of perishables of which 50 per cent were refrigerated. In 1921 these had increased to 19,200 cars, which handled 170,000 carloads of perishables, of which 70 per cent were iced and which required for the purpose 1,600,000 tons of ice. At first a large amount of the ice used in the car bunkers was natural ice, but this was found to be uneconomical and not dependable, so that in 1921 about two-thirds of the ice was manufactured. In 1922 the Pacific Fruit Express Company had a daily capacity of 3500 tons of ice making and storage capacity for 230,000 tons located at 12 ice plants (that is in addition to 5 natural ice plants having 100,000 tons storage capacity),

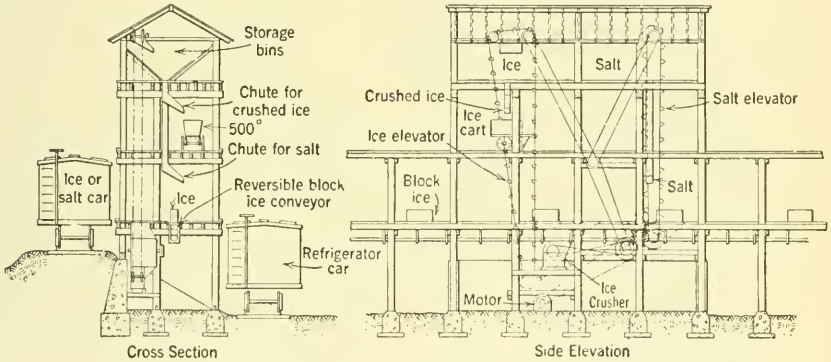



Fig. 309.—Refrigerator Car Icing.

and had 1300 tons daily ice making and 80,000 tons storage capacity under contract with private concerns. The separate plants vary in size, that at Roseville, California, having a daily capacity of 800 tons and a storage capacity of 46,000 tons, and during 1921 using a total of 165,000 tons of ice at this point with a maximum for one day of 2400 tons. The standard icing platform is the island type (Fig. 308) for 55-car trains on each side. The platforms, which are of wooden construction, are 14 ft. high and 14 ft. wide, and the ice is moved by means of an endless chain conveyor. The train is iced during the time of inspection and change of engine and caboose. The procedure in California (1924) in loading a refrigerator car is as follows: the car is cleaned thoroughly and receives its load of ice and salt. It is then sent to the point of shipment, receives its load of perishables, and is brought back with all possible speed to the car icing division point and is made up into trains where enough ice is added to carry it to the next icing point. From

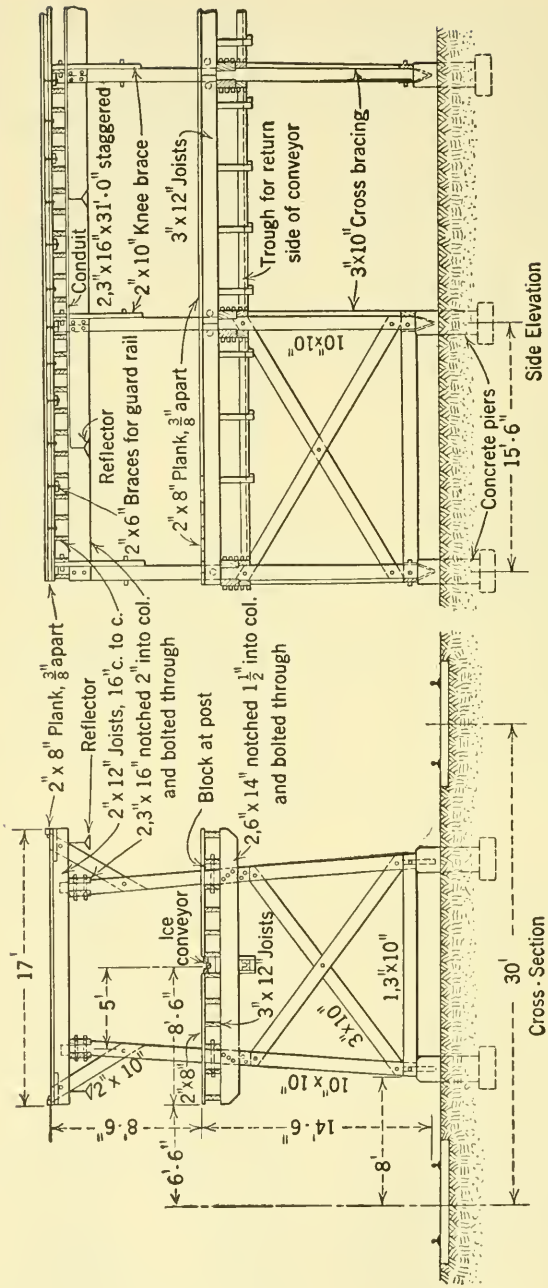



FIG. 310.—Refrigerator Car Icing.

5 to 6 tons of ice are required to ice a freight car. Figures 308 to 311 show ice and salt loading platforms and devices.

**Precooling.**—The development of the precooling industry has been rapid because of the failure in accomplishing a quick chilling of the lading in the ordinary refrigerating car. At the present time (1927) there are three principal means of precooling. First, there is the shippers' precooling design where the lading is placed in cold storage rooms in which the proper temperature is carried and the temperature of the lading is dropped as quickly as possible by the use of fan circulation of the air or otherwise. After the proper temperature has been reached, the lading is loaded into precooled refrigerator cars, having been iced a sufficient length of time previously to cool the car properly. The second means of precooling is the carriers' precooling design. By this is meant that the car is loaded with warm goods after which the entire car and lading are cooled mechanically. Such means can be used only where large amounts of perishable produce (such as citrus fruits) originates, as for example, near San Bernardino, Calif., and Zellah, Wash. In these plants the cars are placed in a precooling dock, the ventilator hatches are connected with large air pipes and the lading is cooled by means of forced circulation of cold air. Finally there is the third method of precooling certain vegetables, including lettuce. Here, water at 35 deg. F.

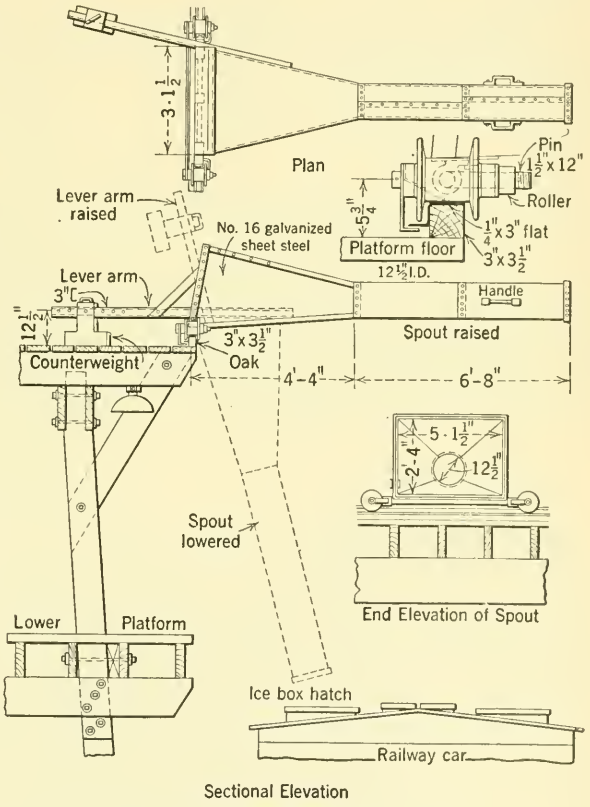



Fig. 311.—Refrigerator Car Icing.

is arranged to flow over the produce and cooling is accomplished very quickly. In the paper by J. W. Andrews[19] the water is cooled by means of Baudalot coolers. The crates of celery are made to pass through a pan of water at a temperature of from 33 to 35 deg. F., and in 20 min. some 70 gal. of water have passed through the crate, after which the surplus water is removed and the celery is delivered to the refrigerator car door at or near 35 degrees temperature. A carload of celery can be cooled and stored in the car in $1\frac{1}{3}$ hours. This means of cooling by direct contact appears to be efficient and economical and seems to improve the product.

*Shippers' Precooling.*—In the state of Florida there were in 1924 15 plants with an annual shipment of 4000 cars of perishables. In this locality salt (NaCl) brine is used for cooling the air, using a spray chamber of about 15 ft. by 9 ft. by 9 ft. high and air velocities of about 200 ft. per min. The usual eliminator removes the brine mist and the air is then passed through overhead ducts designed for 1000 ft. per minute velocity. Reversing of the air flow is provided for usually by reversing the motor drive on the fan. The air is forced into the room (Figs. 312 and 313) where the crates are closely packed, and the air travel is directed by means of baffle curtains "*B*" at the top of the room, resulting in an air pressure drop of from 0.14 to 0.25 in. of water, and there is stimulated a fairly uniform flow of air through the boxes. The inside box temperatures are also given in the figure.

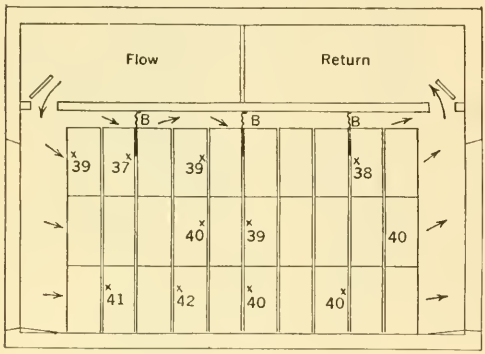



Fig. 312.—Precooling.

The results of tests according to the paper by J. W. Andrews show that the average initial temperature was from 73 to 77 deg. F. Cold air circulation was applied for $10\frac{2}{3}$ hours to get an average temperature of the crates of 40 degrees. The temperature of the air circulated was 36 deg. F.

*The Carriers' Precooler.*—The cooling of the car and the lading with cold air is known best as the Gay process. In his experiments Norman Gay found that all freight cars, even when closed as tightly as possible,

[19] J. W. Andrews, Precooling of Fruits and Vegetables, Refrigerating Engineering 1924.

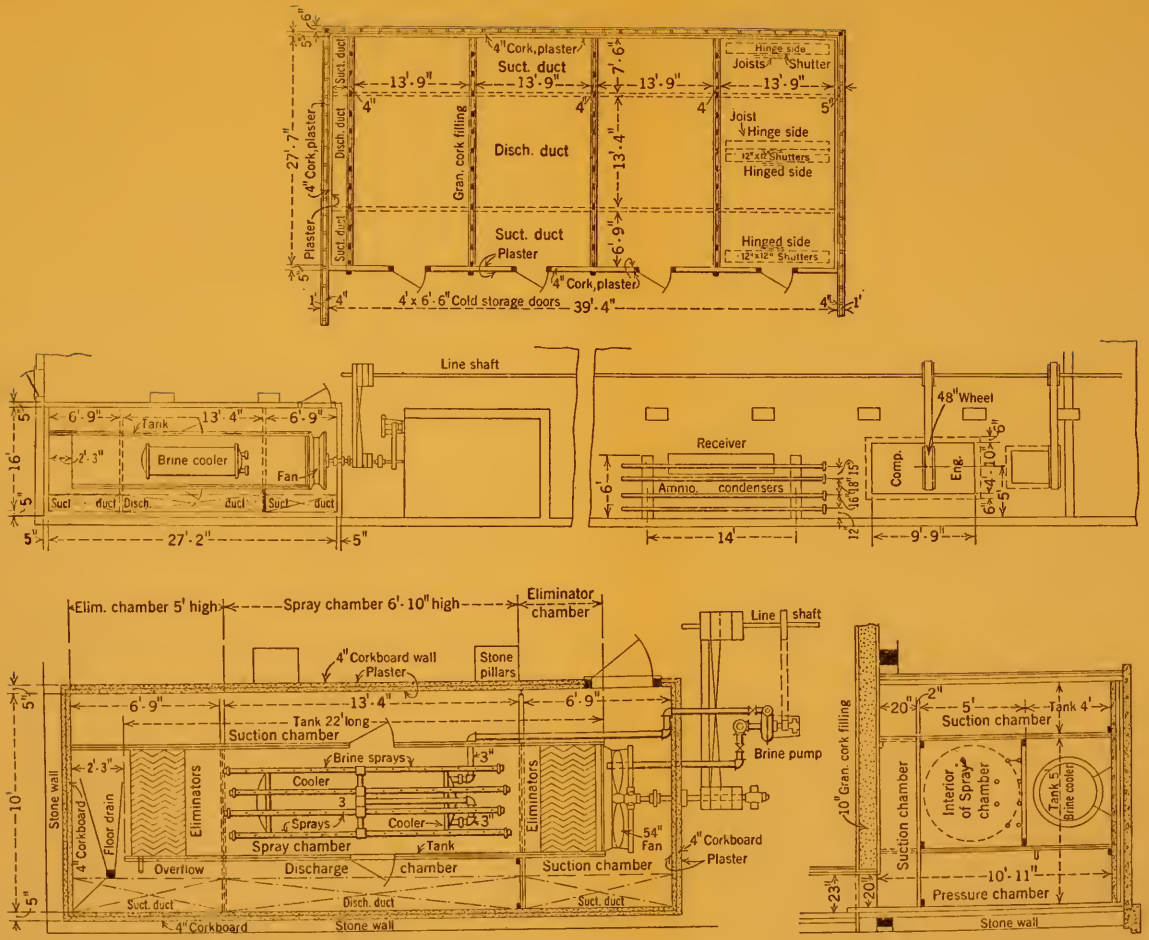



FIG. 313.—Precooling with Brine Cooled Air.

*To face page 488.*

would permit a loss of from 800 to 2000 cu. ft. of air per minute under $\frac{1}{2}$ in. static pressure and the same loss was obtainable with $\frac{1}{2}$ in. vacuum. Therefore any system whereby air is to be forced into the car would need to be " balanced " as far as the pressure in the center of the car is concerned. To do this a pressure fan as well as an exhaust fan is required and adjustment is made so that on opening the center door no blast is noticed in either direction although from 6000 to 8000 cu. ft. of air per minute is being forced through the car. At the San Bernardino plant the lateral flexible pipes had an internal diameter of 22 in., and 32 cars connected together at one time, requiring each 8000 cu. ft. of air per minute, will be cooled within 4 hours at the maximum and fully chilled and iced in 5 hours. The air in the plant was moved by 8 No. 11 Sirocco fans of a total of 300,000 cu. ft. per minute capacity, and this air was cooled in a brine bunker room. The brine itself was cooled by double-pipe brine cooler of 200 tons capacity. The San Bernardino plant stopped the use of the Gay system in 1924.

**Marine Refrigeration.**—Marine refrigeration is an application of the art of cold storage peculiar to itself. It is, of course, restricted by the limitations imposed on all marine engineering design and, in addition, the ship is usually subjected to the condition of one-way traffic—at least in the respect of refrigerated cargoes. The result is that refrigerating vessels need to be designed so as to take general cargoes, and the hatchways must be made large enough for the purpose. In general, a well designed banana carrier (as far as refrigeration is concerned) will be satisfactory also for carrying citrus fruits, but the longer the trips the faster the ship's speed for economical results. However, a ship designed for carrying mixed frozen and chilled meats would not be satisfactory for frozen meats entirely, nor for a general cargo, inasmuch as fruits, for instance, require ventilation.

*Insulation.*—Insulation[20] usually is corkboard, granulated cork, or sometimes mineral wool which may be mixed with granulated cork. When the insulation is all cork, the sides and the overhead consist of an exposed finished layer of wood boards, two layers of 2-in. corkboard, one layer of boards and paper with the space between them, and the ship's side filled with granulated cork.

When mineral wool is used as the main insulation it consists of wool between an exposed finished layer of wood boards and the ship's sides of the deck. For the intermediate and the weather decks the insulation is placed underneath, whilst for the tank top or the floor of the hold it has to be capable of carrying the weight of the cargo. In these latter

[20] A. H. Baer, Journal, American Society of Refrigerating Engineers, 1919.

cases the insulation is usually from 5 to 6 in. of corkboard laid in pitch and with a finishing layer of boards.

Marine refrigeration [21] has four kinds of refrigerating equipment.

1. *Fruit Carriers* need forced indirect air circulation (also external fresh air ventilation). Bananas, for example, are loaded in bulk but are not dunnaged although they are divided by means of bins.

2. Frozen meats require no ventilation but the space is heavily insulated. The cargo is loaded in bulk, suitably dunnaged for natural air ventilation. This cargo should be kept at from 12 to 15 deg. F.

3. Chilled meats require a minimum height in the storage space, which is heavily piped under the ceilings, and the meat to be supported by the deck girders by means of the usual meat hooks running on rails. As temperature changes are not permissible beyond a small limit, the piping controls must be good.

4. This sub-division includes general cargoes. Frozen meats may be stored in the lowest hold. However, this division is a combination of the other three.

Banana carriers require considerably more refrigerating capacity than meats and other cargoes because as a rule the banana is not precooled, but the cargo and the hold should be precooled whenever possible. The ship should be capable of carrying apples and citrus fruits. The temperature required for bananas is 53 to 55 deg., for oranges 42 to 45 deg., for lemons 55 deg., and for apples 32 to 33 deg. F., with particular attention to the ventilation. The cargo should be raised off the decks by means of dunnage battens and (if the fruit is in packages) every alternate row should be vented vertically so as to insure proper air circulation throughout the bulk of the cargo. Bananas are stacked vertically, uncrated, usually two rows high and a third row lying flat unless the voyage is greater than 10 days. The decks are insulated, preferably, on the under side, and care should be taken to prevent construction which interferes with this ventilation. The refrigerating load consists of the heat leakage, the cooling of the cargo and the cooling and the dehumidifying of the fresh air required for ventilation and to keep the carbon dioxide content of the air in apple storage at 10 per cent or less. Figure 314 gives a good idea of the methods used in such work. Bunker rooms appear to be used entirely, contrary to the practice in stationary work, where the water or brine spray is used frequently.

The Food Investigation Board of the British Department of Scientific and Industrial Research has recently published Special Report No. 20, entitled "The Problems of Apple Transport Overseas," which is a gen-

[21] Llewellyn Williams, Naval Arch. and Marine Eng., November, 1924.

eral survey and summary of the results obtained by a scientific expedition to Australia in 1923.

A synopsis of the report is given as follows:

The Australian apple export season of 1922, although a record in point of the number of cases shipped, was in some ways one of the most disappointing in the history of the industry. Of the total export of approximately 2,000,000 cases, it has been estimated that between 500,000 and 600,000 arrived in England with the fruit content in a severely damaged condition. From three of the largest consignments, totalling altogether nearly 400,000 cases, more than 80 per cent of the fruit is said to have been sold as damaged, at a total loss estimated at about £125,000, while the whole season's losses based on the prices of damaged fruit as compared with the

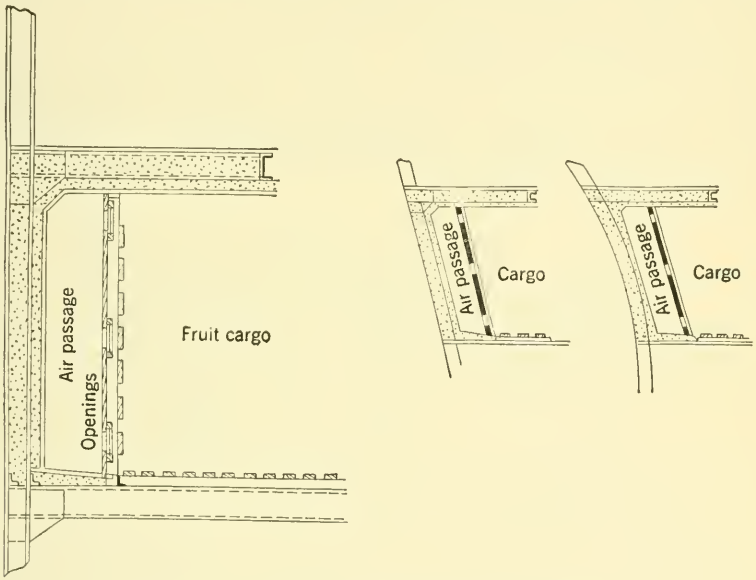



Fig. 314.—Marine Refrigeration—Ventilation.

market value for sound fruit at the same date, have been placed in the neighborhood of £250,000. The figures are taken from a statement prepared by the agent-general for Tasmania.

This type of damage, now know as brown heart, had previously been encountered in the Australian trade on a smaller scale. The nature and cause of the trouble were not, however, known, so that the matter appeared to be one urgently requiring scientific investigation.

The outstanding questions which it was hoped that the expedition would be able to answer conclusively were:

(*a*) Is the disease predetermined some time before its actual appearance, by "inherent vice"?

(*b*) Does brown heart appear before shipment, or in the holds during the voyage?

(*c*) What are the conditions either in the orchard, or during handling and transport which give rise to the disease, and how are they to be avoided?

The evidence obtained by the expedition, together with that already published, thus indicates that brown heart is developed on board ship during the homeward voyage, that the conditions which give rise to it are those accompanying insufficient ventilation of the hold, and that such conditions can be guarded against and all danger of the disease's occurring eliminated.

### The Amount of Ventilation Necessary

The amount of ventilation necessary depends in the first place upon the rate of carbon dioxide formation in the holds. This in turn depends mainly upon:

(*a*) The weight of apples present;

(*b*) The inherent respiratory activity of the apples (varying in different seasons, at different stages of ripeness, and for different varieties);

(*c*) The temperature of the apples.

The second of these factors must be regarded for the present as an unknown variable, the values of which are, however, within known limits. We need only be concerned for the present purpose with the known maximum and minimum values.

In an ideal case, therefore, in which no accidental leakage is occurring, the amount of intentional ventilation by fans required is very roughly as follows:

| Cargo, Temperature, Degrees F. | Percentage of Carbon Dioxide Maintained in the Hold | Ventilation, Cubic Feet of Air per Day per Ton of Apples |
|---|---|---|
| 70 | 10 | 120 |
| 70 | 1 | 1200 |
| 35 | 10 | 25 |
| 35 | 1 | 250 |

In atmosphere control there are two alternatives. In the first place, safety—that is the avoidance of brown heart—may alone be aimed at. In that case, it would appear that holds with air circulation systems are on the whole satisfactory as at present operated. With the grid system, however, safety cannot be insured without gas-registering instruments and provision for ventilation.

The second and more scientific alternative is to maintain deliberately a desired carbon dioxide percentage level, with the purpose of taking advantage of the retarding effect of this gas on the ripening of the fruit. From this point of view, the amount of ventilation given by the air circulation systems would appear to be excessive, and the grid system without fans or trunks would seem to offer greater possibilities for atmosphere control. By restricting ventilation as far as is compatible with safety, the amount of snow on the grids is kept as low as possible, since ventilation implies the admission of outside air, and, unless this be artificially dried, it brings with it a large amount of moisture.

No definite statement regarding the amount of this intentional ventilation can be made to cover all cases, except that it should be such as to maintain the carbon dioxide concentration in the hold at not more than 10 per cent. In some cases, no

intentional ventilation may be necessary, the accidental leakage alone being sufficient. In other cases, it appears that as much as 10,000 cu. ft. of the hold atmosphere per day may have to be replaced by fresh air. The carbon dioxide percentage in the hold is the indicator by which the ship's engineer must regulate ventilation, and it would therefore be essential to have instruments of carbon dioxide in every hold. Atmosphere control without such instruments is as impossible as temperature control without thermometers.

### Problem of Temperature Control.

It is well known that the length of life of apples in storage is prolonged more or less proportionately by a reduction in temperature. This is due to two causes:

1. A retardation of the rate of ripening—the low temperature causing a slowing up of all vital processes.

2. The checking of mold development—the spores of these fungi, which are invariably present on the skin of the fruit, being largely prevented from germinating and invading the tissues.

The desiderata with regard to temperature are therefore:

1. Cooling of the fruit to 32 deg. F. to 34 deg. F. with the least possible delay after gathering.

2. The maintenance of a uniform temperature of 32 deg. F. to 34 deg. F. throughout the bulk of the cargo, and throughout the voyage.

The results obtained on the expedition show that it is necessary to recognize the great difficulties which have to be overcome in order to obtain even a fair working approximation to these desired conditions, and that the best means of overcoming these difficulties is not yet known. The difficulties increase in proportion to the bulk of the cargo carried, and the size of the hold employed. To-day, refrigerated cargo holds, carrying apples, may have a capacity of 80,000 cubic feet, and this space is filled with an almost solid block of apple cases.

Calculations based on data obtained by the expedition indicate that the amount of heat which has to be removed in cooling this cargo initially, say from 70 deg. F. to a carrying temperature of 35 deg. F., is about as great as the total amount of heat subsequently to be removed during a voyage of average length. The heat being generated at any time by the apples (due to respiration) is on the average three or four times as great as the heat leaking in through the insulated walls and floor from the sea. The heat which refrigeration has to remove therefore comes at all times largely from the cargo, which not only brings in a large amount of solar heat when loaded warm, but also continuously generates heat throughout its mass during the voyage.

The information required was obtained to some extent by the indirect method of constructing three-dimensional temperature maps of the hold, at different times, throughout the voyage. This method has the additional advantage of yielding direct results as to the extreme range of temperature distribution over the hold (in departure from uniformity), after the cargo has been cooled down, and has reached a steady state.

The results obtained by the expedition show that in large holds none of the systems in use attain a uniform temperature throughout the cargo after cooling. The temperature in different parts of the hold varies rather widely even after cooling down is completed and the steady condition of storage reached. There seems little doubt that slight modification in practice will effect considerable improvement in

this respect. Each of the different systems, however, presents its own particular technical problem—a problem of detail. In general, the most important agents in promoting both quick cooling and uniform temperature appear to be (1) roof grids and vertical dunnage or (2) high velocity, alternating, transverse, cold air draughts with transverse dunnage.

*Chilled Meat.*—Chilled meat is suspended from meat rails from the under side of the decks, and the best deck height in the clear is 6 ft. 6 in., although 9 ft. 6 in. can be used and even 12 ft. 6 in. or 15 ft. 6 in. below the rail has been successful, but extremes are bad on account of poor air circulation and the difficulty in stowing. Temperatures may be specified by the shipper, but $28\frac{1}{2}$ to $29\frac{1}{2}$ is a good value. As a rule, 35 days is the maximum storage for successful operation. Refrigeration is obtained by means of brine pipe coils suspended from under the ceiling and on

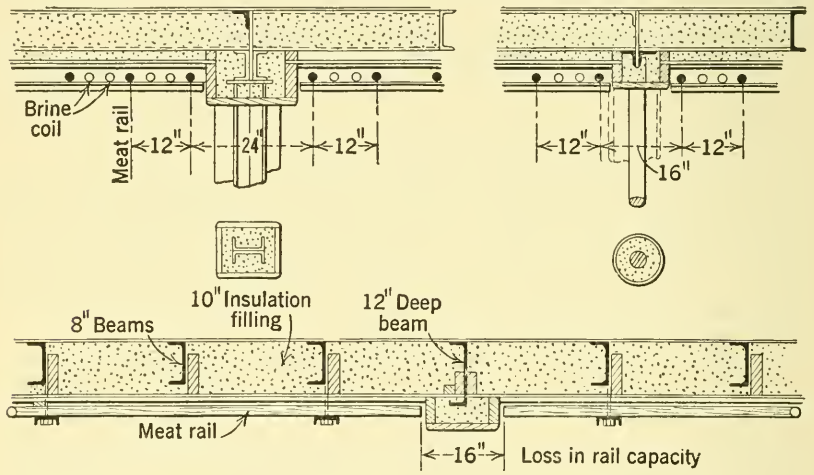



Fig. 315.—Marine Refrigeration—Piping, Insulation and Meat Rails.

the sides of the bulkheads, arranged preferably on the so-called interlaced system, with the cold, tepid and warm brine available with an attemperator (Fig. 318) for the proper control of the temperature. Too cold a temperature will cause a shrinkage of the meat, and usually a temperature of the brine of 15 to 18 deg. F. is good practice. The cargo is packed tight enough to prevent undue swaying and yet permit adequate air circulation. Bulkheads and side coils must be protected from the meat by means of battens. Figure 315 gives a good arrangement of rails and piping.

*The General Carrier.*—The general carrier must have space for all usual commodities, and be arranged to provide two temperatures.

**Mechanical Equipment.**—Lloyd's requirements specify sufficient tonnage to maintain the necessary temperature with 18 hours' operation. The capacity of the refrigerating machine using carbon dioxide while operating in the tropics is some 60 per cent of the capacity in the North Atlantic waters, whereas the ammonia machine is 80 per cent. Figure 316 gives an idea of the ratio of the storage space to the tonnage required, allowing for the reserve specified by Lloyd's, under nominal speed of the compressor. The kind of refrigerant used on shipboard is not mandatory. If ammonia is used it must be in a separate room and vented (usually on one of the upper decks), whereas carbon dioxide may be used in the main engine room. Some of the trade uses ammonia almost entirely, as, for example, the British ships in the River Plate

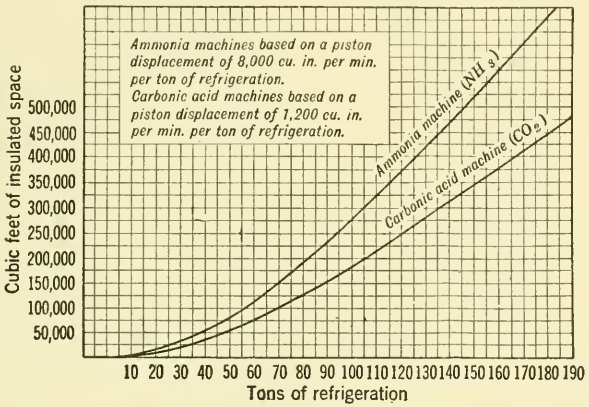

Ammonia machines based on a piston displacement of 8,000 cu. in. per min. per ton of refrigeration.
Carbonic acid machines based on a piston displacement of 1,200 cu. in. per min. per ton of refrigeration.
Ammonia machine ($NH_3$)
Carbonic acid machine ($CO_2$)
Cubic feet of insulated space
500,000
450,000
400,000
350,000
300,000
250,000
200,000
150,000
100,000
50,000
10 20 30 40 50 60 70 80 90 100 110 120 130 140 150 160 170 180 190
Tons of refrigeration

Fig. 316.—Marine Refrigeration—Tonnage Required of Carbonic and Ammonia Compressors.

service, and some of the largest refrigerator ships on the Australian and the New Zealand routes. The brine cooler room contains the brine tank, the distributing header and the pump connections. Insulation is usually of 4 in. corkboard in two layers, or its equivalent. The brine cooler is usually the double pipe or the shell and tube for ammonia, whereas with carbon dioxide the continuously welded iron pipe coils in separate nests in a cast iron box, or continuous circular iron pipe coils also nested are usual.

The condensers are of copper for carbon dioxide, of the submerged design, nested in rectangular boxes in the frame of the compressor. As copper cannot be used for ammonia, the double-pipe ammonia condenser as well as the shell and tube are mostly used. All equipment must be paired, or duplex, and sufficient spares must be carried in accordance with Lloyd's regulations.

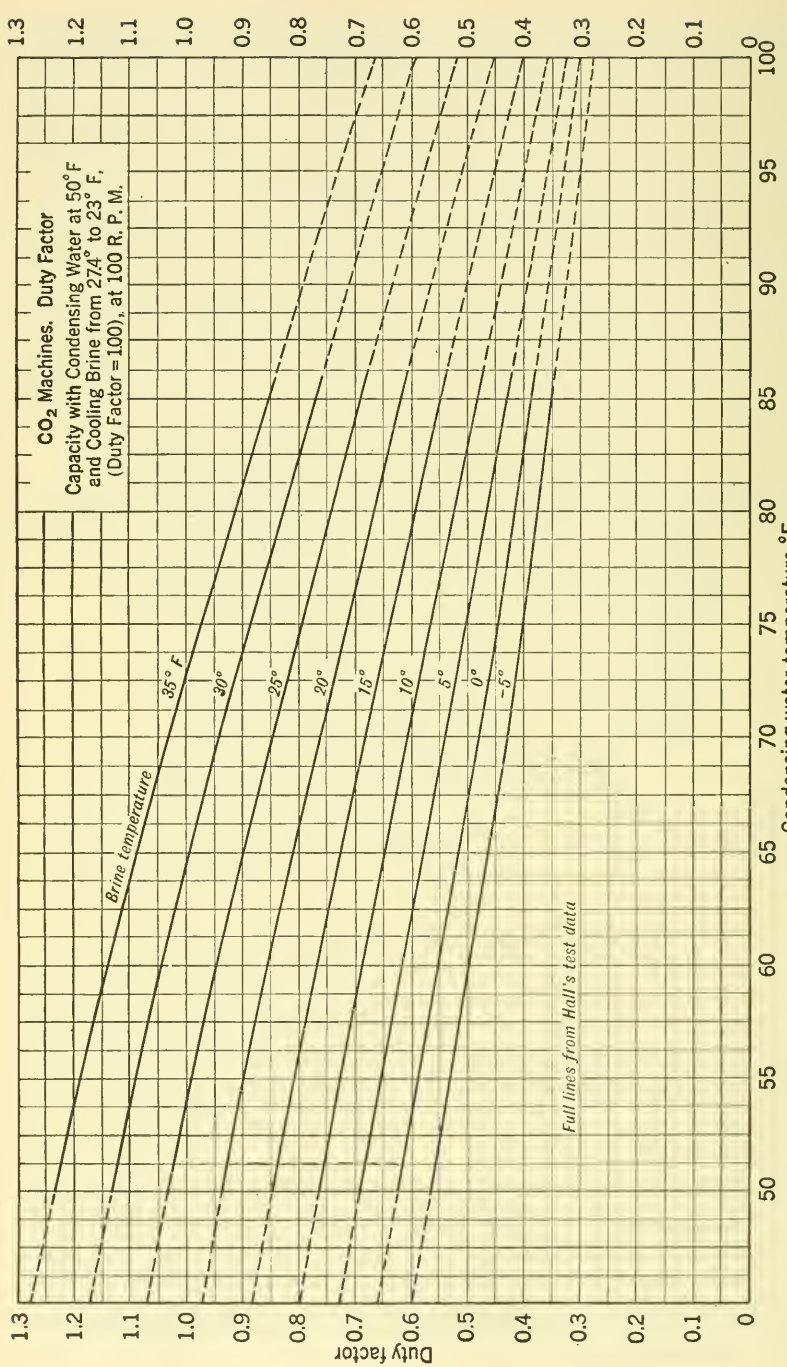
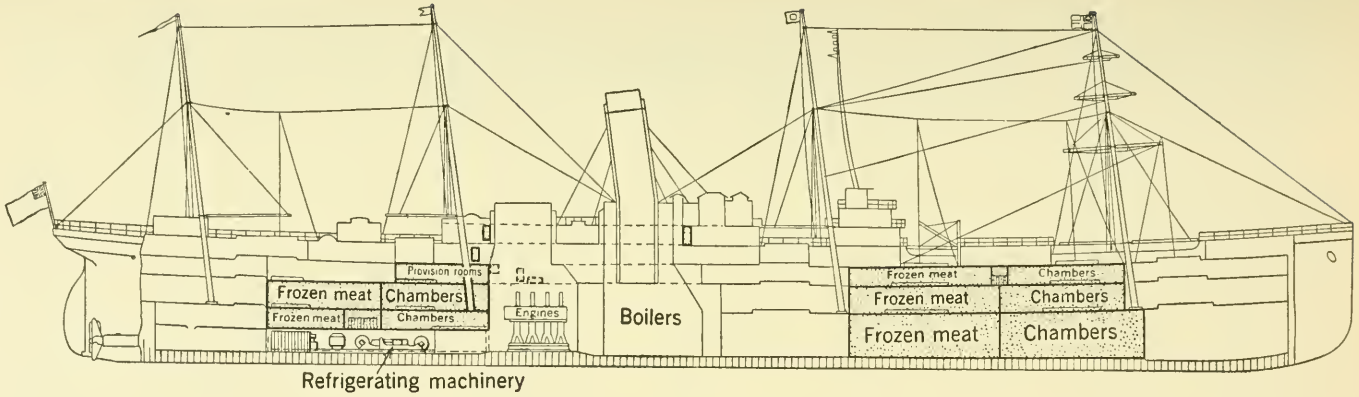
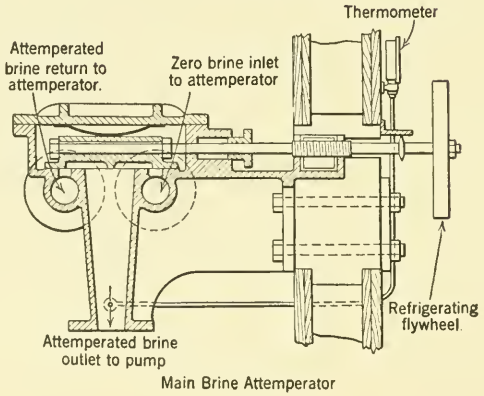



FIG. 317.—Effect of Condensing Water Temperature and Evaporating Temperature on Duty of Carbonic Compressors.



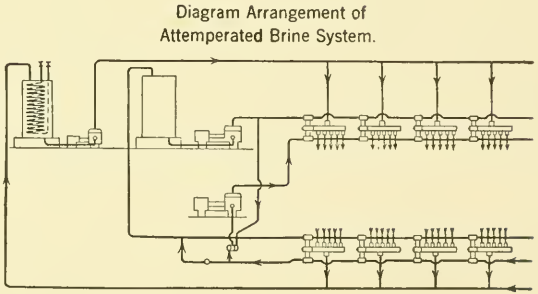




FIG. 318.—Marine Refrigeration Details.

### DISTRICT COOLING (Pipe Line Refrigeration)

Pipe line refrigeration has, like central heating systems, particular and special applications. Its field is in the rather congested commission house and cold storage warehouse district, where the feed lines do not have to be made excessively long in order to pick up the load. The service needs to be fairly well established at the beginning as extensions cannot be made with economy after the initial installation, except to spread out into a wider area by installing a new compression plant at a new central location. The result has been that applications of pipe line refrigeration are relatively few in number and consist of plants located in the larger cities such as Boston, New York, Philadelphia, St. Louis, Kansas City, etc. The Quincy Market Cold Storage and Warehouse Company, of Boston, has changed from direct expansion to brine, and is now carrying over 1400 services with 1500 tons refrigeration of connected load, but the warehouse service is separate. The Merchants Refrigerating Company, of New York, has brine refrigeration only (at zero deg. F.), and carries 400 services from 200 to 15,000 cu. ft. of cold storage and besides has some small warehouses aggregating 2,750,000 cu. ft., of which 35 per cent is freezer and 65 per cent is cooler space (31 to 35 deg. F.). The St. Louis plant, however, is direct expansion, and in general it can be said that the brine system construction costs from 40 to 60 per cent more than does the ammonia direct expansion.

The choice as to whether a brine system is to be used or a direct expansion one, depends to a certain extent on local conditions and cannot be decided offhand without knowledge of all the facts. The pipe line system of refrigeration is in reality only a large ordinary plant where the mains are under the street. The construction cost is in favor of the direct expansion, especially when it is remembered that the liquid and the gas return lines to the compressor are not insulated. This is because of the fact that the evaporating coils in the cold storage boxes are very liberally piped, and the expansion valve is operated so that superheated ammonia is brought back to the street mains by having several boxes in series if necessary and thereby eliminating frost on the pipe lines in the street. The result of this procedure is that the return gas has the temperature of the pipe trench which usually in midsummer (as it is below the frost line) is at 70 to 75 deg. F., so that about 50 degrees of superheat is contained in the return gas. If direct expansion is employed the piping is usually designed for some 10 to 15 lb. per sq. in. drop from the most remote box to the compressor. If operation requires a boiling temperature corresponding to 25 lb. gage pressure, namely 12 deg. F., then approximately 10 lb. gage pressure will be had at the

compressor. The loss in capacity (as compared with 25 lb. gage and 10 degrees superheat) is 48 per cent, and the increased horse power per ton of refrigeration is 73 per cent. The loss of ammonia in the Boston plant before the change to brine operation is stated to have been as much as $15,000 to $20,000 per year.

When using brine both supply and return lines need to be carefully insulated. Judging from the experiences of the Boston and the New York systems, the larger plants are practically obliged to use brine. The control of the temperatures in the boxes is made possible very easily by the regulation of the amount of the brine flowing through the box coils, and there is no danger from the escaping fumes—should a break in the pipes occur—either to life or to the commodity being stored. With brine it is possible to install indicating and control systems, which will indicate by an alarm when leaks in the brine line develop, of the magnitude of approximately one-half barrel of brine within a determined short time interval, and on hearing this alarm the operator can close within 30 seconds all the main stop valves in the street system by closing the switch connected to the motors on these valves.[22] By this arrangement the brine system is divided into a number of parts, the leak is localized, and the remainder of the brine system can be put back on the load, without much loss of brine and without waiting for repairs to be made. Continuity of service is easier with brine than with direct expansion by utilizing brine storage, as, for example, the Merchants Refrigerating Company (operating with zero degree brine), has a large amount of strong calcium chloride brine kept at 32 deg. F., whereas the Boston plant circulates 10,000,000 gal. of brine per 24 hours and has 6,000,000 gal. in reserve. This plant makes use of a high-duty reciprocating pumping engine.

As regards the cost of the operation of the brine system, this is, of course, dependent on the design. The problem is very similar to that of a cold storage warehouse. The circulating pump may have only the friction head and the velocity head to overcome in the case of balanced systems, and it may have in addition the lift to the highest point in the piping if the system is an open one. The pressures against which the pump has to operate are given by A. W. Oakley as 60.5, 55.5, and 50.5 lb. per sq. in., corresponding to 113.5, 104.2, and 95.0 ft. of brine head respectively. Using a nominal value of 6.0 gal. of brine per ton of refrigeration per minute, and a density of 1.23, the horse power per ton becomes

$$\frac{6.0 \times 1.23 \times 8.33 \times 113.5}{33{,}000} = 0.211.$$

[22] A. W. Oakley, Refrigerating Engineering, December, 1924.

**Piping.**—The piping for brine can be full weight steel or wrought-iron pipe, using flanged joints and ring gaskets and in some cases the alternate joint is made with heavy cast-iron sleeves and is caulked with lead, or this may be of cast iron using the bell and spigot, the flanged and screw thread or the gland end types. For ammonia it is usual to employ full weight pipe for pipe sizes 4 in. and over and extra heavy pipe for sizes under 4 in. The Merchants Refrigerating Company uses a dual system, so that if one circuit develops a leak the other circuit can continue with the load until the leak is repaired. The direct-expansion ammonia system requires besides the liquid and the suction gas line a third line for pumping out purposes. It is necessary to place all piping below the frost line (from 3 to 7 ft. below the surface), and if possible to confine all connections to the manholes which are located at the intersecting streets. The tendency seems to be to use all welded pipe as far as possible between manholes. Expansion joints must be provided every 500 ft. at a maximum, and every 300 ft. where heavy distributing pipe lines are taken off the mains. These expansion joints should be located at the manholes, and they may be of the slip joint, the corrugated or the U-type, the pipe being suitably anchored at some convenient point thereby permitting the change of length of the pipe to be taken up by the expansion joint. The pipe is subjected to a possible change of temperature of from 70 to 100 deg. F. The change of pipe length corresponding to this temperature change is 0.75 in. per 100 ft. of pipe. If the slip joint type is used the finished surface of the sleeve should be kept smeared with grease so as to prevent tearing of the packing of the gland which may under extreme conditions have a movement of of 2.5 to 3.0 in. Valves not placed in the manholes should have an extension valve stem. Service branches should not come up into the frost lines, and should take off from the top of the main, using a nipple and ell so as to permit some freedom of movement to conform to the expansion and the contraction of the main. The coils in the boxes should all have stop valves in the supply and the return, and (for brine) air valves at the high points and a lock valve so as to regulate the maximum amount of brine flow. The brine cooler generally used is the shell and tube design, since this is efficient and easily cleaned. The number of passes of the brine varies considerably, and there may be two or more up to (possibly) eight, but as a rule four passes of the brine through the tubes are sufficient.

**Insulation.**—The type of insulation varies with the different installations. In Fig. 319 there are several methods in use. If a wooden box is used the wood should be waterproofed (kananized treated). The pipe conduit or box is packed, frequently, with a concrete of granulated cork and pitch.

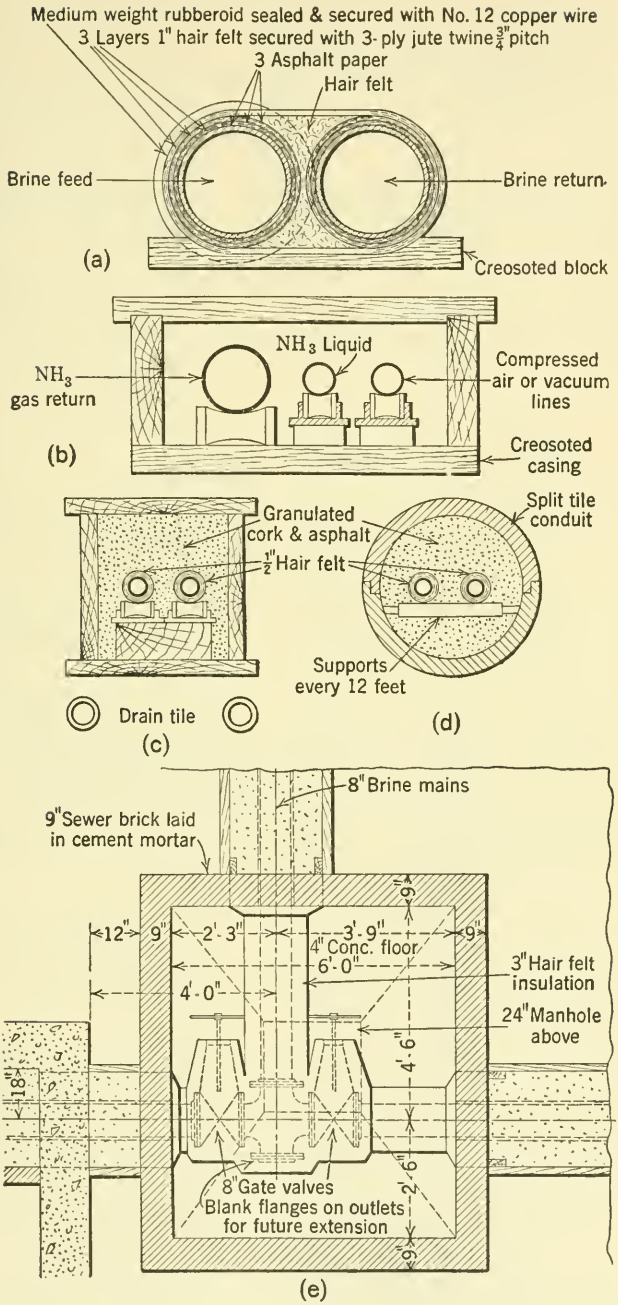



FIG. 319.—District Cooling Details of Piping.

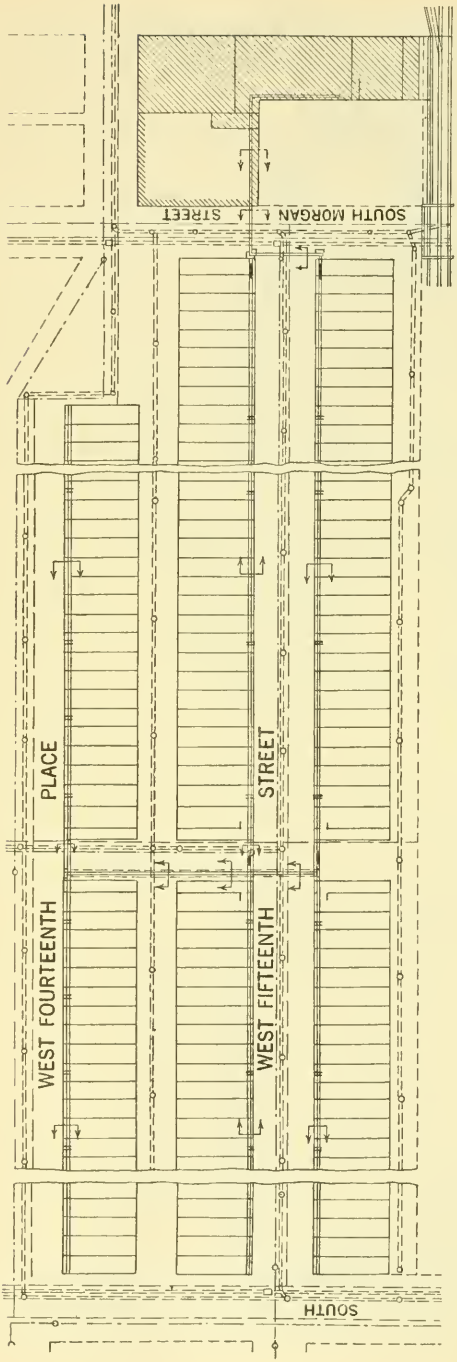



FIG. 320.—District Cooling—Street Piping.

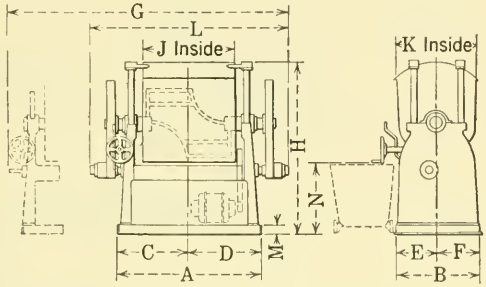
R. H. Tait[23] says that in St. Louis 59 per cent of all connections are of 300 cu. ft. capacity or less, 17 per cent are of 300 to 600 cu. ft., 14 per cent are of 600 to 1200 cu. ft., and 10 per cent are of 1200 and over, and that he figured on 7500 cu. ft. of cold storage at 32 to 36 deg. F. per ton of refrigeration, whereas the practice in Boston, according to F. L. Fairbanks, is only 2800 cu. ft. The load on the plant depends on the relative amounts of freezer and cooler space, as the freezer is refrigerated for from 9 to 12 months, whereas the cooler is under load only about 6 months. The larger boxes are usually charged according to the refrigeration supplied, but the smaller ones are on a flat rate per cubic foot, depending on the details of the box and the temperature carried in the room.

## REFRIGERATION IN THE BAKERY

The modern bakery of 100,000 loaves per week includes in its process

[23] R. H. Tait, American Society of Refrigerating Engineers, 1913.

several steps where mechanical refrigeration is now considered an economic necessity. These include (*a*) the 50 deg. F. cold storage room of about 100 to 200 sq. ft. floor area for yeast, shortening, malt and milk; (*b*) a cooling tank for lowering the temperature of the dough water to 40 degrees, or lower, on the basis of about 25 per cent water by weight; (*c*) the cooling of the mixers either by the use of brine at 10 to 20 deg. F. or water coils with water at about 35 degrees with the addition of crushed ice. The water coil system is the popular method of cooling, but blowing the air at about 32 degrees is increasing in use; (*d*) cool the air in the dough room (mixer) down to 78 degrees when this is necessary and (*e*) provide a cold storage for waxed paper at 55 to 60 deg. F. to prevent the paper sticking.[24]




| BBL | A | B | C | D | E | F | G | H | J | K | L | M | N | H.P. | CLUTCH SHAFT SPEED |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1½ | 51" | 32" | 25½" | 25½" | 16" | 16" | 5' 9" | 5' 8⅜" | 27" | 26" | 6' 3" | 4¼" | 28⅛" | 3 | 243 |
| 2 | 51" | 32" | 25½" | 25½" | 16" | 16" | 5' 9" | 5' 8⅜" | 27⅜" | 26⅜" | 6' 3" | 4¼" | 28½" | 3 | 243 |
| 2½ | 58" | 35" | 29" | 29" | 17½" | 17½" | 6' 4" | 5' 11" | 38" | 31" | 6' 10" | 4½" | 28½" | 5 | 238 |
| 3 | 58" | 35" | 29" | 29" | 17½" | 17½" | 6' 4" | 5' 11" | 34" | 32" | 6' 10" | 4½" | 28½" | 5 | 236 |

FIG. 321.—Bakery Refrigeration—Dough Mixer.

In mixing the dough it is desired to develop the gluten so that the structure of the dough will be sufficiently elastic and strong enough to hold in confinement the gas which is produced by the fermentation process. The power which is consumed in the mixing develops heat (the heat equivalent of the work done) and the more rapid the mix the faster the temperature rise will take place. A certain type of mixer requires 20 minutes for the mix, and every minute over 10 minutes gives a marked increase in the quality, but should heating prevent the last part of the process an inferior bread will result. The cooling by blowing cold air into the dough as it is mixed has a bleaching effect and produces a white bread, and it appears to assist fermentation.[25]

[24] W. W. Reece, Chicago Section, A.S.R.E., October, 1923.

[25] H. L. Fisher, A.S.R.E., April, 1923.

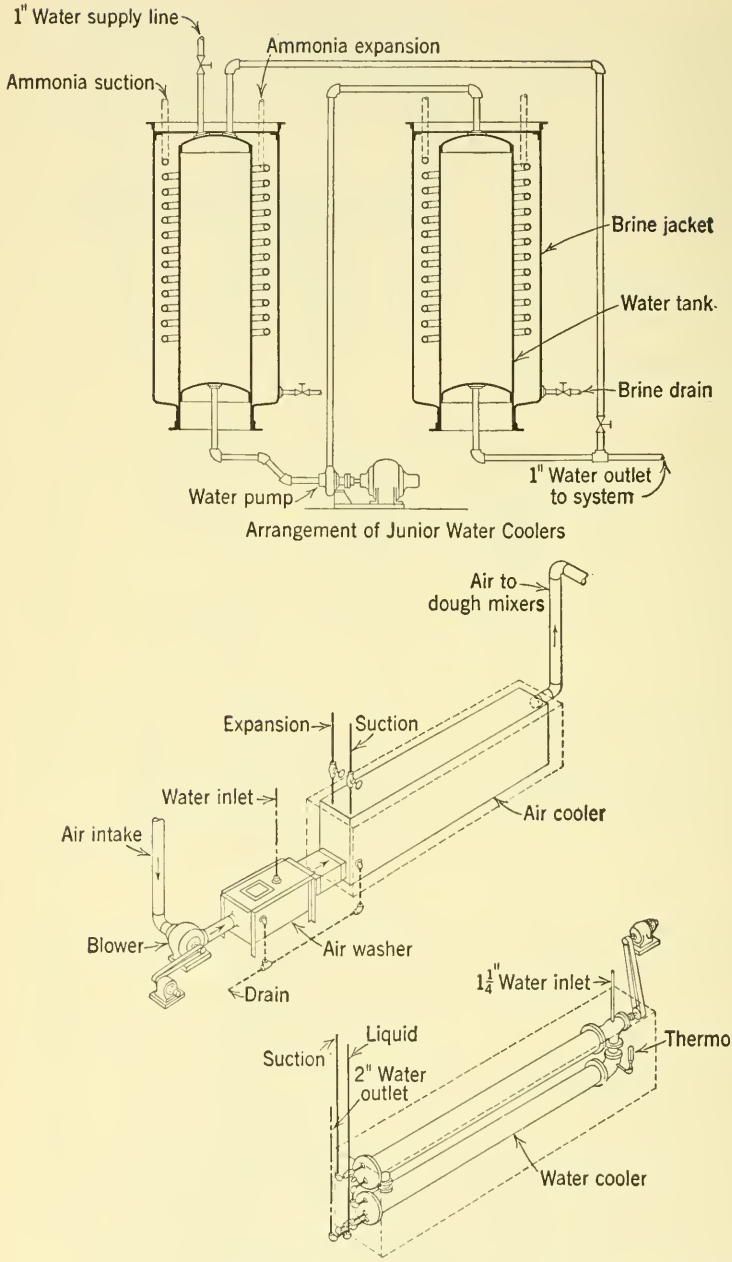



Fig. 322.—Bakery Refrigeration—Water and Air Coolers.

In bakery refrigeration it is usual to figure on 250 loaves of bread per barrel of flour of 196 lb. The dough mixers vary in size (Fig. 320), but the 3-barrel type is the one which operates at greatest advantage Such a size would use 588 lb. of flour and 110 lb. of water, cooled to 35 degrees. During mixing, 750 cu. ft. per min. of air at 32 deg. F. and under $3\frac{1}{2}$ oz. pressure is supplied each mixer. The problem of the calculation of the refrigeration requirements varies with the location (as regards the maximum summer temperatures) and on the construction of the building, the exposure, the window areas and other pertinent

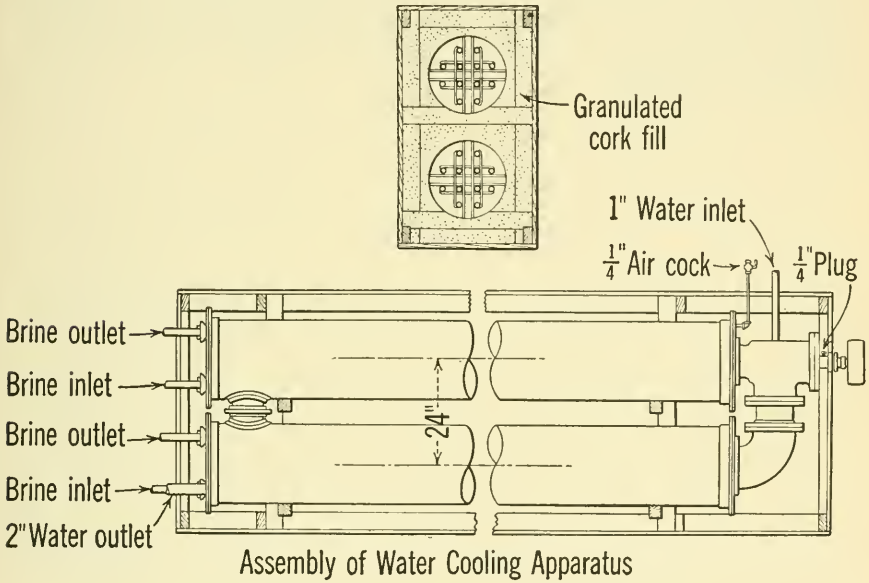



Fig. 323.—Bakery Refrigeration—Water Coolers.

factors. As an illustration, the following will give an idea of the larger refrigerating quantities:

**Problem.**—Find the refrigeration required in a bakery of 60,000 loaves capacity per 8 hours per day. The outside air temperature in 90 and the wet bulb is 75 deg. F.

The capacity is 60,000 ÷ 250 = 240 bbl. per day = 30 bbl. per hr. At the rate of 20 min. per batch there will be an average of 30 ÷ 9 = $3\frac{1}{3}$ mixers in continuous service. The air requirements will be: $3\frac{1}{3} \times 750 = 2500$ cu. ft. per minute to be cooled from a wet bulb temperature of 75 degrees to 32 degrees. The volume of 1 lb. of air is 12.47.

$$2500 \div 12.47 \times (37.8 - 11.8)^{26} = 5220 \text{ B.t.u.}$$

[26] In all air cooling the psychrometric chart (Fig. 326) may be used.

Assuming that the initial water temperature is 80 deg. F., the heat removed is:

$$\frac{110 \times 3\frac{1}{3} \times (80 - 35)}{20} = 825 \text{ B.t.u.}$$

Total refrigeration = 6045 B.t.u.
= 30.2 tons.

As the preceding calculation is liberal it will provide for the small requirements of the storage rooms for the waxed paper, the yeast, shortening, malt and the milk. In fact, plants of this capacity in and near the vicinity of Milwaukee use about 20 tons of refrigeration, but possibly with a longer day than eight hours, in which case the calculated capacity would be reduced in proportion. Figures 321, 322 and 323 give details of bakery refrigeration.

## CHAPTER XV

## THE COOLING AND CONDITIONING OF AIR

One of the most important applications of mechanical refrigeration is that of the cooling and conditioning of air, which, being a fluid, can be made to act conveniently as a carrier of refrigeration. Examples of air cooling and conditioning are: in the process of the chilling of glue, cooling films for cameras and for motion pictures, or even for making gelatine capsules, either because the emulsion needs to be hardened quickly or because the commodity must be chilled quickly from the fluid to the semi-fluid or solid condition. At times the problem becomes one of simply keeping the workroom moderately cool and properly humid, or it may be (as in certain chemical plants) that of lowering in temperature sufficiently the containers of a gas being handled to prevent the danger of its escape into the atmosphere of the workroom. An example of this last is to be found in the case of the shell-filling stations where phosgene gas is charged into shells. Just what system of air cooling needs to be employed in each case depends on the particular process. In extreme cases what is required particularly is a certain low humidity of the air, obtained by cooling the air down to a temperature sufficient to give the desired water content to the air. Such a condition the Gayley method, now obsolete, tried to meet by condensing out the water content of the air by cooling it down to 25 deg. F. (it being desired to hold the moisture content at 20 grains of water per one pound of dry air), subsequently heating it to the usual blast temperature of 3000 deg. F. before supplying it to the blast furnace.

In cooling air it is possible to use a bunker room through which the air is forced and brought into contact with the refrigerated pipes, but this method is poor because of the likelihood of heavy frostation on the pipes and the consequent lowering of the efficiency of heat transfer. The more usual method at present is to cool by the use of a spray chamber, using a water spray in the case of temperatures above 35 deg. F., and water and brine sprays in two separate stages in cases of temperatures below 35 degrees. When water is used it is usually cooled by the use of a Baudalot cooler (Fig. 342), where the refrigerant boils in the inside of the pipes and the water trickles from the outside of the top pipe down to the lowest pipes by gravity. The air to be cooled is circulated through an air chamber where it comes into intimate contact

with the water which is atomized by the nozzles through which the water passes.

**Moisture Content.**—In air cooling and conditioning certain terms, peculiar to the conditions, are used. The *wet bulb* temperature is that temperature obtained when a thermometer with a wetted gauze is exposed to the air in motion—preferably at a velocity of 15 ft. per second. It is a depressed temperature, below that of the ordinary dry bulb temperature, and the amount of this depression depends on the moisture content in the air. The maximum amount of water content which can be absorbed by a unit weight of dry air at constant pressure depends on the air temperature, and this varies from 44.2 grains at 45 deg. F. to

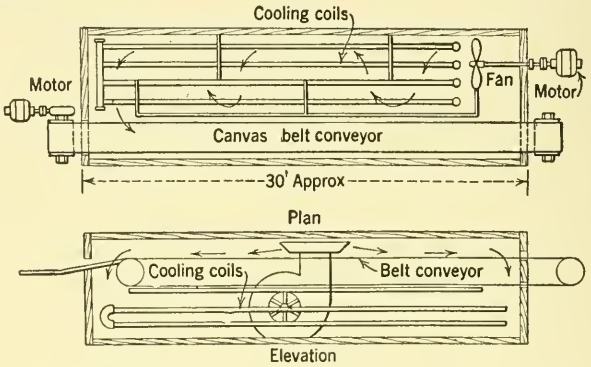



Fig. 324.—Method Employed in Cooling Certain Commodities.

155.8 grains at 80 deg. F. per 1 lb. of dry air. The "*dew point*" temperature is that temperature at which air is saturated with moisture and beyond which any further reduction of temperature will cause a precipitation of water.[1] Incidentally, the dew point and the wet bulb temper-

[1] The humidity may be expressed approximately as a ratio of the pressures. If $p_1$ is the pressure of the water vapor at some given condition, $p_2$ the pressure of saturation at a lower temperature, and $p_3$ the pressure of saturated vapor at the temperature corresponding to the given condition $p_1$, it can be stated approximately that

$$p_1 = p_2 = MBT;$$
$$p_3 = M_1BT;$$

therefore

$$\frac{M}{M_1} = \frac{p_2}{p_3},$$

where $V_1 = 1.0$;
$M$ = the weight in lb.;
$B$ = the gas constant;
$T$ = the absolute temperature in deg. F. abs.

or the humidity is the ratio of the dew point pressure to the pressure of saturation at the temperature of the given condition.

atures are quite distinct. The *relative humidity* can be defined as the ratio of the weight of the moisture present in a certain volume of air to the amount that could be present if the air were saturated. This value is usually expressed as a per cent.

**The Psychrometric Chart.**—Fig. 325[2] represents a chart used for solving problems in air cooling. In the diagram it will be noticed that such a point as "*A*" represents a condition of 65 deg. F. dry bulb temperature and 60 per cent relative humidity. Lines sloping downward to the right are lines of constant wet bulb temperature, and "*C*" is the temperature (56.7 deg. F.) of the wet bulb and "*B*" is the dew point temperature (50.7 deg. F.). The moisture content of "*A*" is 55 grains per one pound of dry air, as is every other point on a horizontal line drawn through "*A*." In the diagram the *total heat* measured from 0 deg. F. is given by the curve (marked "total heat"). It has been found by experiment that the heat removed in the cooling of air can be obtained by getting the difference in the total heats correspond to the wet bulb temperature at the beginning and the final conditions. For example, if air is to be cooled from a condition of 65 deg. F. dry bulb and 60 per cent relative humidity to the dew point temperature, the total heats from the diagram are 24.1 and 20.75 B.t.u. respectively. The volume of a pound of dry air with various moisture contents can be found by referring to the two volume curves; for example, at 51 deg. F. the volume of one pound of dry air is 12.87 cu. ft., and when saturated with moisture it is 13.03 cu. ft. With the foregoing it is now possible to proceed with an illustrative example, and this will be made general enough so that the method of solution may be used in other problems if the details of the particular job are known. In calculating the refrigeration required it is generally necessary to make allowance for all sources of heat, i.e., the heat equivalent of the work done, the heating effect of the electrical losses (the $I^2r$ losses), all process heating, the heat generated by the illumination, animal heat, heat of chemical reactions, etc.

**Problem.**—A room in an industrial plant is to be maintained at a temperature of 65 deg. F. and 60 per cent relative humidity (Fig. 327). Its cubical contents is 204,300 cu. ft., and it has 1248 sq. ft. of glass and 908 sq. ft. of wall surface exposed to the sun's rays, and 2154 sq. ft. of glass, 1863 sq. ft. of wall, 2728 sq. ft. of partition and 12,770 sq. ft. of floor area subject to heat leakage. The outside temperature of the air will be taken as 95 deg. F. maximum, with a wet bulb temperature of 76 deg. F. The infiltration loss will be assumed as a complete change of the air of the room in five hours. The maximum number of operators will be taken as 500 persons at any one time, and the illumination will be taken as 39 250-watt lamps. It is required to find the total refrigeration, the size of the fan required to

[2] Willis Carrier, Transactions of the American Society of Mechanical Engineers, 1911.

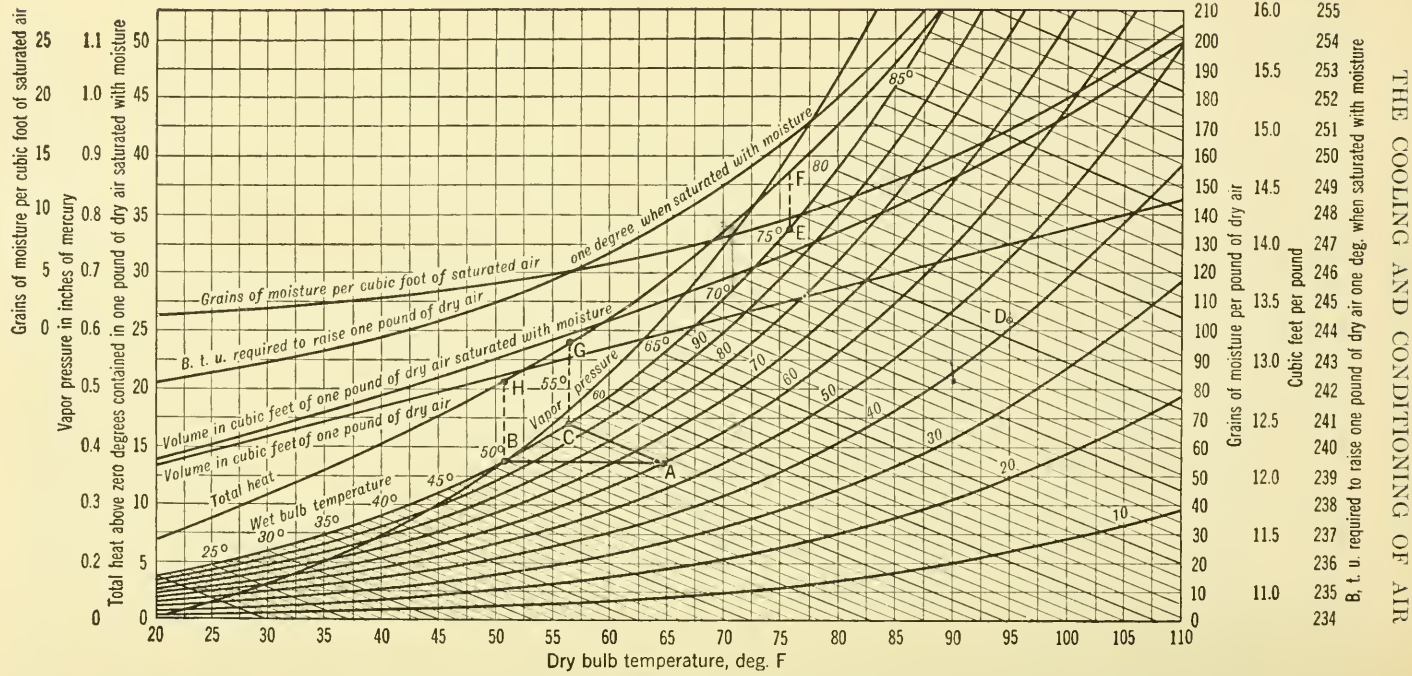

Grains of moisture per cubic foot of saturated air
Vapor pressure in inches of mercury
Total heat above zero degrees contained in one pound of dry air saturated with moisture
Grains of moisture per pound of dry air
Cubic feet per pound
B. t. u. required to raise one pound of dry air saturated with moisture one degree
Grains of moisture per cubic foot of saturated air
B. t. u. required to raise one pound of dry air
Volume in cubic feet of one pound of dry air saturated with moisture
Volume in cubic feet of one pound of dry air
Total heat
Wet bulb temperature
Vapor pressure
Dry bulb temperature, deg. F
A
B
C
D
E
F
G
H

FIG. 325.—The Psychrometric Chart—Method of Using.

circulate the air and the amount of water required to be pumped through the sprays, if the water is to be heated by the air from 43 to 51 deg. F. Also it is required to find the number of sprays, the amount of baffles and eliminator surface, and the size of the ducts to convey the cooled air to the different parts of the room.

*Calculation.*—Referring to Fig. 325, point *A* is found at the intersection of the 60 per cent humidity line and the dry bulb temperature of 65 deg. F. The line *AB* is drawn so as to show the process of heating or cooling at a constant water content in the air—in this case a constant weight of 55 grains of water per pound of dry air. Point *B* indicates that the dew point required from the conditions of the problem must be 50.7 deg. F., and the wet bulb temperature (point *C*) must be 56.7 degrees. If the air passed through the spray chamber is to have the *sensible* heat only removed from it so that the moisture content will always remain at 55 grains, then the air

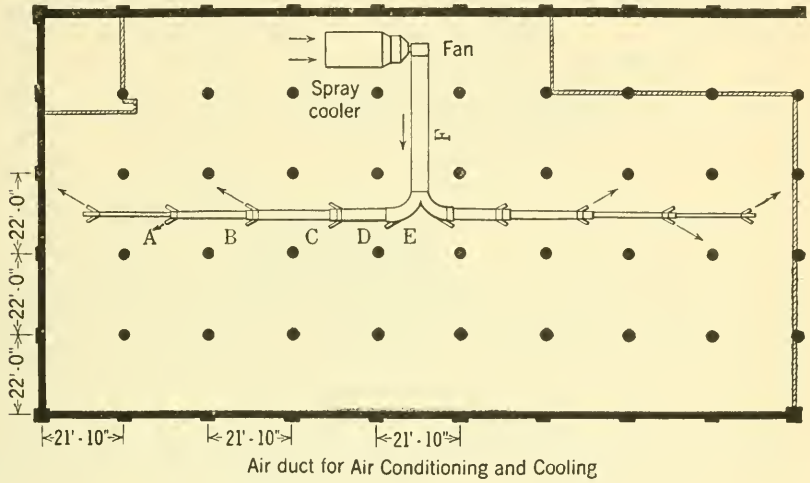



FIG. 327.—Problem in Air Conditioning.

must be cooled down to 50.7 deg. F. and sufficient air must be circulated in order that the total heat entering the room, and tending to raise the air temperature, will be removed by the sprays.

*Heat Leakage.*—The heat leakage will be:

*The effect of the direct radiation* (of the sun):

1248 sq. ft. of glass at 0.5 B.t.u. per sq. ft. (per minute) = 624 B.t.u. per minute.
908 sq. ft. of wall at 0.1 B.t.u. per sq. ft. (per minute) = 91 B.t.u. per minute.

(These values of 0.5 and 0.1 are practical experimental values to account for the radiant heat of the sun's rays.)

*People.* $500 \times 5.0 = 2500$ B.t.u.

(The value 5.0 is the *sensible* heat given off by each workman per minute. This and the other values in the calculation of the amount of air to be circulated by the fan are the heat units tending to raise the temperature of the air.)

*Heat Leakage* (by conduction) [3]:

| | | |
|---|---|---|
| *Glass.* | $2{,}152 \times 0.46 \times (95 - 65)$ | = 29,700 B.t.u. per hour. |
| *Wall.* | $1{,}863 \times 0.075 \times (95 - 65)$ | = 4,200 B.t.u. per hour. |
| *Partition.* | $2{,}728 \times 0.09 \times (95 - 65)$ | = 7,370 B.t.u. per hour. |
| *Floor.* | $12{,}770 \times 0.08 \times (95 - 65)$ | = 30,700 B.t.u. per hour. |
| | Total heat leakage per hour | = 71,970 B.t.u. |
| | Total heat leakage per minute | = 1,200 B.t.u. |
| *Infiltration.* | $\frac{204{,}300 \times 0.250 \times (95 - 65)}{60 \times 5.0 \times 13.5}$ | = 379 B.t.u. per minute. |
| *Illumination.* | $\frac{39 \times 250 \times 42.4}{746}$ | = 556 B.t.u. per minute. |
| | Total sensible heat per minute | = 5,350 B.t.u. |
| | 5 per cent for contingencies | = 267 B.t.u. |
| | Estimated heating effect of the circulating fan | = 272 B.t.u. |
| | Total | = 5,889 B.t.u. per minute. |

This amount of sensible heat, 5889 B.t.u., is the total heat entering the room from all the sources tending to raise the temperature of the air in the room, and this is the only factor affecting the capacity of the fan. As refrigeration is to be accomplished by means of water sprays in a spray chamber it is necessary to move *enough* air through this chamber so that heat will be absorbed by the water at the rate of 5883 B.t.u. per minute. This air so circulated leaves the room at 65 deg. F. and reenters it again from the sprays at 50.7 deg. F. In other words there is a rise of temperature of 14.3 degrees, and the ability of the air to absorb heat will depend on this rise of temperature and its specific heat. If the specific heat is taken as 0.245 B.t.u., then the number of pounds of air to be circulated per minute will be

$$\frac{5889}{14.3 \times 0.245} = 1680 \text{ lb.}$$

As the air entering the fan is saturated with moisture the specific volume will be 13.03 cu. ft., and the total volume required at the entrance of the fan will be

$$1680 \times 13.03 = 21{,}890 \text{ cu. ft. per minute.}$$

*Refrigeration Required.*—The preceding calculation was only for the purpose of estimating the size of the fan. The heat absorbed by the air is *not* the refrigeration required of the refrigeration plant. As a matter of fact it is only about one-half of the required amount. In the present problem a certain amount of air has to be renewed, and a certain amount may be recirculated. In this problem 10 per cent will be fresh air, and 90 per cent will be recirculated, and will be cooled from the conditions of 60 per cent humidity and 65 deg. F. dry bulb to 50.7 degrees. The fresh air will be cooled from 95 deg. F. dry bulb and 76 wet bulb to the same temperature of 50.7 degrees. The total heat in either case will be found by going upwards

[3] These values can be found by the method given in Chapter VI.

from the wet bulb temperature, 100 per cent humidity line, to the total heat line and then reading the total heat from the corresponding point on the scale.

| | |
|---|---|
| The total heat at 76 deg. wet bulb | 38.7 B.t.u. |
| The total heat at 50.7 deg. wet bulb | 20.75 B.t.u. |
| Heat removed per 1 lb. air | 17.95 B.t.u. |

The total cooling of the fresh air will be $0.10 \times 1680 \times 17.95 = 3020$ B.t.u.

| | |
|---|---|
| The total heat of the air at 56.7 deg. F. wet bulb | 24.20 B.t.u. |
| The total heat of the air at 50.7 deg. F. wet bulb | 20.75 B.t.u. |
| Heat removed per 1 lb. of air | 3.45 B.t.u. |

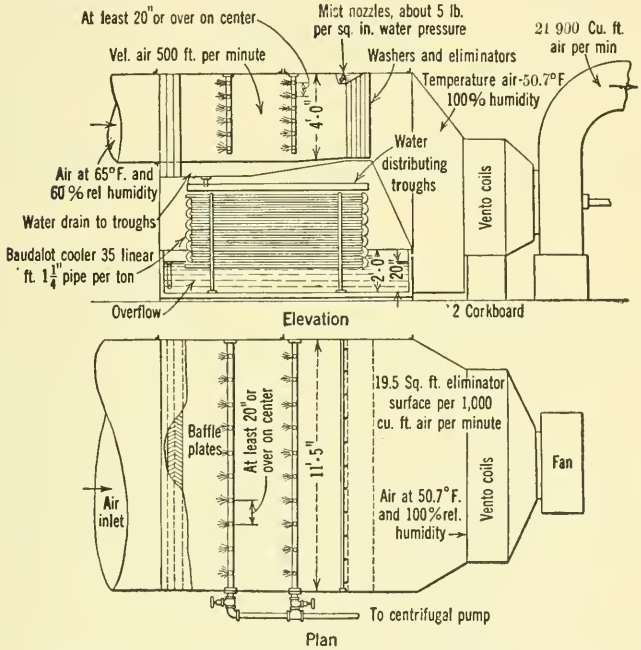

At least 20" or over on center
Mist nozzles, about 5 lb. per sq. in. water pressure
21 900 Cu. ft. air per min
Vel. air 500 ft. per minute
Washers and eliminators
Temperature air-50.7°F. 100% humidity
4'-0"
Water distributing troughs
Air at 65°F. and 60% rel humidity
Water drain to troughs
Vento coils
Baudalot cooler 35 linear ft. 1¼" pipe per ton
2'-0"
20"
Overflow
2 Corkboard
Elevation
19.5 Sq. ft. eliminator surface per 1,000 cu. ft. air per minute
Baffle plates
At least 20" or over on center
11'-5"
Fan
Air inlet
Air at 50.7°F. and 100% rel. humidity
Vento coils
To centrifugal pump
Plan

FIG. 328.—Water Spray Chamber for Air Conditioning.

The total cooling of the recirculated air is $0.90 \times 1680 \times 3.45 = 5210$ B.t.u. A certain amount of moisture will enter the air in the room, which will tend to increase the moisture content. This will be condensed in the spray chamber, and will add this much more to the refrigerating load. The amount of this moisture entering with the air of infiltration is

$$\frac{204{,}300}{5 \times 60} = 681 \text{ cu. ft. per minute.}$$

The volume of 1 lb. of air at 95 deg. F. and 42 per cent relative humidity is equal to $13.98 + (0.82 \times 0.42) = 14.32$ cu. ft. When this air enters the room it will be

cooled to 65 degrees and some of the water content will be condensed. The excess moisture brought into the room will be then:

$$\frac{681 \times (108 - 55)}{14.32 \times 7000} = 0.360 \text{ lb.}$$

As only 90 per cent of the air in the room is recirculated the refrigeration required is

| | | |
|---|---|---|
| | $0.90 \times 0.360 \times 1060 =$ | 344 B.t.u. |
| To these must be added: | | |
| The latent heat of the operators, | $0.9 \times 500 \times 3.0 =$ | 1,350 B.t.u. |
| The heat effect of the water circulating pump (estimated) at 5 hp. | = | 212 B.t.u. |
| Radiation effect into the room (estimated) | = | 50 B.t.u. |
| Total refrigeration | = | 10,186 B.t.u. |
| 5 per cent for safety | = | 509 B.t.u. |
| | | 10,695 B.t.u. |
| | | or 53.50 tons |

The water in the spray chamber is heated from 43 to 51 degrees and the number of pounds of water required will be

$$10{,}695 \div (8.3 \times 8.0) = 161 \text{ gallons per minute,}$$

and the circulating system should be designed for at least 180 gallons per minute so as to carry any overload for a short time should occasion demand it.

*The Spray Chamber.*—Having found the refrigerating load for the room to be 53.5 tons, the amount of air to be cooled to 50.7 deg. F. and circulated to be 21,900 cu. ft. per minute and the amount of water to be pumped through the sprays to satisfy the conditions of the problem to be 180 gal. per minute, the next step is to determine the amount of surface required in the spray chamber and the sizes of the various pieces of apparatus to be used in the spray chamber. First, however, it should be said that with the temperatures required in this type of problem water is always used for cooling the air, and a form of spray nozzle is used that will atomize the water. Cooling coils in the spray chamber may or may not be used (kept wet by means of flooding nozzles) in order to assist in the cooling of the air. In this design the sprays only will be used, and the eliminator plates (provided to remove the surplus water carried in the air) will be provided with special flooding nozzles in order to keep them clean, and to increase their efficiency. The smaller air cooling plants may have a Baudalot water cooler directly underneath the spray chamber so arranged as to be flooded by the water passing out of the spray chamber, in a manner similar to the method used with atmospheric condensers receiving water from cooling towers and spray nozzles. The objection to this method where ammonia is used is that there is no particular safety from bad leaks, although the ammonia passing out through a small leak would be absorbed by the water for a time and there would be some warning of the fact of the leak. This method will be used in the present design on account of its simplicity, but it is well to keep in mind that the use of brine might easily be imperative at times. However, with the use of extra heavy pipe, first class fittings and reasonable care and inspection during erection and operation there should not be any appreciable danger (Fig. 328).

*Number of Spray Nozzles.*—Referring to the table of the capacities of spray nozzles (Table 101), it will be seen that with the $\frac{3}{16}$-in. diameter nozzle opening and a pressure of 20 lb. per sq. in., 15.0 lb. of water per minute will be sprayed, or $15 \div 8.3 = 1.8$ gal. per minute. The problem requires 161 gal. per minute at the very minimum, and 180 gal. should be allowed for. There will be required then:

$$180 \div 1.8 = 100 \text{ nozzles under a pressure of 20 lb. or}$$

$$180 \div \frac{17}{8.3} = 88 \text{ nozzles under a pressure of 25 lb.}$$

These will be arranged in two banks, each of 50 nozzles, spaced not more than 20 in. apart, and not less than 6 in. between centers in order that a satisfactory "curtain" of cold water is provided for the air to pass through. As the velocity of the air in this part of the system must be not more than 500 ft. per minute, the cross-sectional area becomes $22{,}000 \div 500 = 44.0$ sq. ft. If the head room limits the height of the chamber proper to 4 ft. 0 in., then the width becomes 11ft. 0 in. Assuming that the nozzles are to be 12 in. on centers horizontally, then 10 stands will be required of 5 nozzles high. Evidently some arrangement, say using 10 stands spaced 13 in. on centers horizontally and 5 nozzles high on $7\frac{1}{2}$ in. centers will give satisfactory spacing.

TABLE 101

SPRAY NOZZLES

Capacities in Pounds of Water Discharged per Minute under Varying Pressures
(Carrier Engineering Corporation)

| Diameter of Pipe Connections, Inches | Diameter of Openings, Cap. Inches | Pressures in Pounds per Square Inch | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 10 | 20 | 25 | 30 | 40 | 50 | 60 | 70 | 80 | 90 | 100 |
| $\frac{1}{4}$ | $\frac{1}{32}$ | .27 | .39 | .44 | .48 | .55 | .62 | .68 | .73 | .79 | .83 | .89 |
| $\frac{1}{4}$ | $\frac{1}{16}$ | 1.0 | 1.5 | 1.7 | 1.8 | 2.1 | 2.4 | 2.6 | 2.8 | 2.9 | 3.2 | 3.4 |
| $\frac{1}{4}$ | $\frac{3}{32}$ | 2.3 | 3.3 | 3.7 | 4.0 | 4.6 | 5.2 | 5.7 | 6.1 | 6.5 | 6.9 | 7.3 |
| $\frac{3}{8}$ | $\frac{1}{8}$ | 4.6 | 6.5 | 7.3 | 8.0 | 9.2 | 10.0 | 11.0 | 12.0 | 13.0 | 14.0 | 15.0 |
| $\frac{3}{8}$ | $\frac{3}{16}$ | 10.0 | 15.0 | 17.0 | 18.0 | 21.0 | 23.0 | 26.0 | 28.0 | 29.0 | 31.0 | 33.0 |
| $\frac{1}{2}$ | $\frac{1}{4}$ | 18.0 | 26.0 | 29.0 | 32.0 | 37.0 | 41.0 | 45.0 | 49.0 | 52.0 | 56.0 | 58.0 |
| $\frac{3}{4}$ | $\frac{3}{8}$ | 51.0 | 72.0 | 81.0 | 88.0 | 102.0 | 114.0 | 125.0 | 135.0 | 144.0 | 153.0 | 162.0 |
| 1 | $\frac{1}{2}$ | 91.0 | 128.0 | 144.0 | 157.0 | 181.0 | 202.0 | 222.0 | 240.0 | 257.0 | 272.0 | |
| $1\frac{1}{4}$ | $\frac{5}{8}$ | 127.0 | 182.0 | 204.0 | 226.0 | 257.0 | 287.0 | 310.0 | | | | |
| $1\frac{1}{2}$ | $\frac{3}{4}$ | 186.0 | 263.0 | 295.0 | 332.0 | 370.0 | 415.0 | | | | | |
| 2 | $\frac{7}{8}$ | 252.0 | 356.0 | 400.0 | 435.0 | 501.0 | | | | | | |
| 2 | 1 | 330.0 | 465.0 | 522.0 | 570.0 | 657.0 | | | | | | |
| $2\frac{1}{2}$ | $1\frac{1}{4}$ | 515.0 | 728.0 | 815.0 | 890.0 | | | | | | | |

1 Cu. ft. of water at 4 deg. C. weighs 62.428 lb.
1 gal. of water weighs 8.3 lb. = 231 cu. in.

*Eliminators.*—If the baffle plate construction is used in order to reduce the eddy currents, give a uniform velocity to the air, and prevent water from splashing out

of the spray chamber, it is customary to use just enough to rectify the air (Fig. 335). The eliminator surface is calculated on the basis of $19\frac{1}{2}$ sq. ft. (calculated for one side only) of surface per 1000 cu. ft. of air per minute or $(22{,}000 \div 1000) \times 19.5 = 430$ sq. ft. of surface is wanted. The *Carrier* eliminator construction is one with crimped gutters set vertically on $1\frac{1}{8}$-in. centers, each with six corrugations at 60-degree directional change of the air, each corrugation about $2\frac{5}{8}$ in. wide, and arranged so that the first three will have smooth sides and corners and the last three will have stamped lips for catching the entrained moisture.

The *Webster* eliminator is designed for four corrugations, with 30-degree deflection of the air, made up of two plates: the rear staggered with respect to the front, the separate plates spaced about 3 in. on centers, and 5 in. wide. The Webster practice is to discharge the water spray in the direction of the air flow, and to place them not less than 4 ft. 3 in. in front of the eliminators. These sprays use about 20 lb. per sq. in. pressure at the nozzle, and in addition "mist" nozzles are used at 5 lb. pressure for flooding the eliminator surfaces.

The *American Blower Company* uses about 40 sq. ft. of scrubber surface per 1000 cu. ft. of air per minute, spaced 1 in. with 35-degree deflection of the air. These are made with plates having three corrugations and a final stamped lip. The eliminator surfaces are in three rows, the final row spaced $3\frac{1}{4}$ in., and designed on the basis of 12 sq. ft. of surface per 100 cu. ft. of air per minute. The spray nozzles are designed to give a uniformly distributed spray across the spray chamber, and the spray is in the direction of the air flow.

Using the Carrier design, the eliminator surface, spaced on $1\frac{1}{8}$-in. centers, becomes:

$$(132 \div 1\tfrac{1}{8}) \times 6 \times 2.63 \times 4 \div 12 = 617 \text{ sq. ft.}$$

The required amount is 427 sq. ft., or there is about 44 per cent excess area over the requirements.

*The Baudalot Cooler.*—The figure shows the design with the water cooler directly below the spray cooler chamber. In cooling water great care must be exercised to safeguard the pipes against freezing, and so the Baudalot cooler is used instead of the double-pipe cooler or the shell and tube type which would be preferred usually when cooling a non-freezing liquid. Such Baudalot coolers are usually designed on the basis of 35 lin. ft. of $1\frac{1}{4}$-in. pipe per ton of refrigeration. The problem requires a refrigeration of 10,560 B.t.u. per minute, or 52.8 tons. The surface, then, is

$$35 \times 52.8 = 1850 \text{ ft. of } 1\tfrac{1}{4}\text{-in. pipe.}$$

It is doubtful whether a cooler more than 8 ft. long could be arranged for with satisfaction in this type of design, so using 14 pipes high and pipes 8 ft. long the number of stands becomes $1850 \div (8 \times 14) = 16.5$, and this can be arranged for easily on 6- or 7-in. centers. Figure 328 gives a good idea of the entire spray design.

The problem has been worked out for the simplest design, i.e., a self-contained Baudalot cooler for the water, using direct expansion in these pipes for the ammonia; the entire spray chamber being located inside the room to be cooled. Without question the spray chamber could be placed in any convenient location and the Baudalot cooler could have been cooled with the use of carbon dioxide or brine could have been pumped in from an adjacent building.

*Cold Air Ventilating Ducts.*—In general, for processes using the spray chamber, such as gelatine or other industrial work, it is desired to keep a product exposed to

cool air and yet to air dry enough to prevent the surface of the product from having moisture condense on it. This means the selection of the proper operating temperature, relative humidity and corresponding dew point temperature, as has been outlined already. It is necessary to supply the chilled air as uniformly as possible in order that the room temperature will be uniform throughout. This is best done by some duct system, as indicated by Fig. 327. The problem requires the circulating of 22,000 cu. ft. of air per minute at a dew point temperature of 50.7 deg. F. The problem consists of finding the duct diameters at the different sections and the total head required of the fan to give the desired circulation.

In order to calculate the friction head due to the flow of air in ducts it is convenient to use a chart as is shown in Fig. 330, showing the resistance to the flow of air at a standard temperature and pressure at different velocities of flow in the pipe. The general formula for the loss of head due to the flow of a gas in pipes and ducts is:

$$h_a = f\frac{LP}{A} \times \frac{V^2}{2g},$$

where

$h_a$ = loss of head due to flow in feet of the medium flowing (air in this case);
$h$ = loss of head in inches of water;
$f$ = coefficient of friction;
$L$ = length of pipe or duct, in feet;
$P$ = perimeter of pipe or duct, in feet;
$A$ = cross-sectional area of pipe or duct, in feet.
$\frac{V^2}{2g}$ = velocity head;
$V$ = average velocity, in feet per second;
$d_a$ = density of air (0.075), lb. per cubic foot;
$d_w$ = density of water (62.4 lb.), per cubic foot;
$g$ = acceleration due to gravity.

As it is customary to state the head in inches of water, the formula may be expressed as:

$$h = h_a \times \frac{12 \times d_a}{d_w} = f\frac{12 d_a LPV^2}{d_w A 2g},$$

which is simply an expression for the ratio of the densities of the two substances. Therefore the second expression gives the loss of head in inches of water. Using an average value for $f$ of 0.0036 for air velocities of from 1000 to 3000 ft. per minute, and with $L$ = 100 ft., the formula becomes

$$h = \frac{12 \times 0.075 \times 100 \times 0.0036}{62.4 \times 64.32} \times \frac{P}{A} V^2,$$

and for round ducts $P/A = 4/D$, and therefore

$$h = 0.000323\frac{V^2}{D}$$

(where $D$ is the diameter of the duct, in feet).

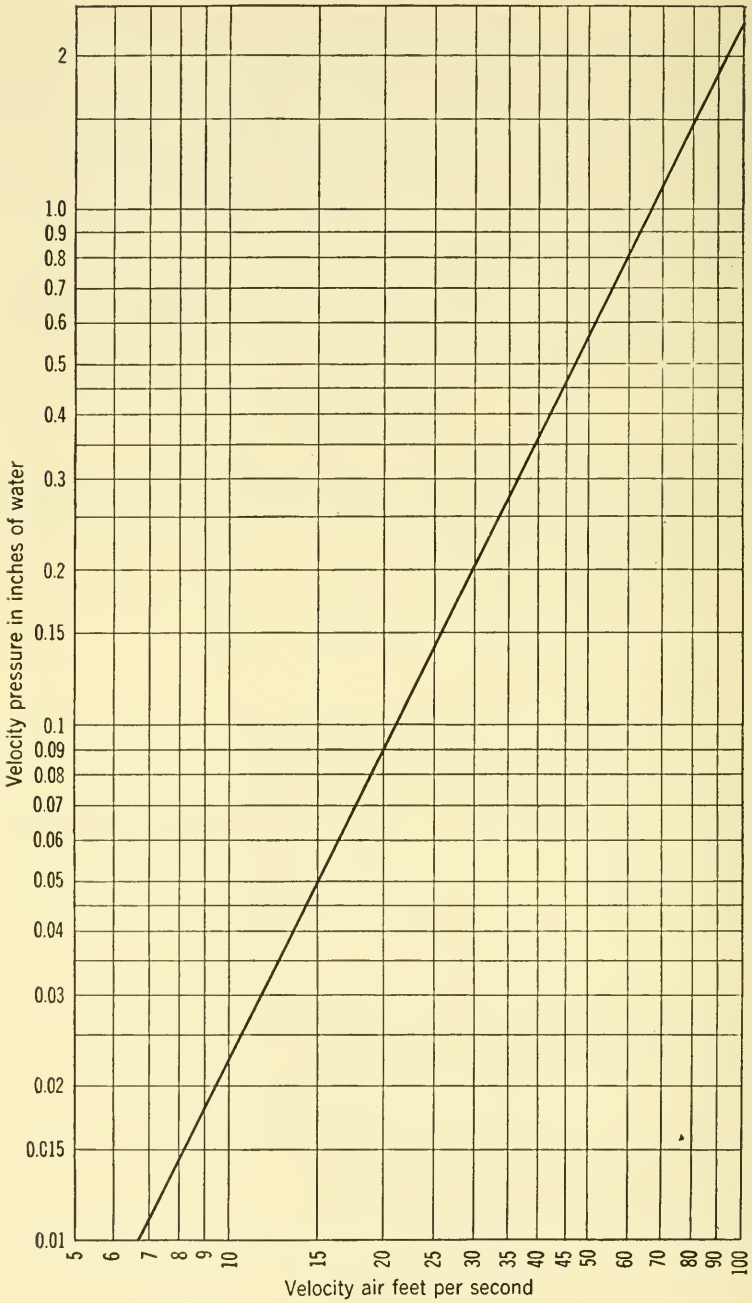



Fig. 329.—Velocity Pressure Head for Air.

If $Q$ is the cubic feet per minute of air flowing, then $v = \frac{Q}{60A}$, and, making this substitution, the formula finally becomes

$$Q = 6678h^{\frac{1}{2}}\left(\frac{A^3}{P}\right)^{\frac{1}{2}},$$

and, if the duct is round, then

$$Q = 2625h^{\frac{1}{2}}d^{\frac{5}{2}} \quad \text{(for a duct 100 ft. long, and where } f = 0.0036\text{)}.$$

In the figure it will be noted that the quantity of air (in cubic feet per minute) is represented by the horizontal lines and that the resistance to the flow in inches of water per 100 ft of length is given by the vertical lines, whereas the different sizes of round ducts or the width of duct for rectangular cross-sections is given by the lines sloping upwards to the right.

In all work dealing with the flow of fluids, the terms total, static, and velocity are used. The static pressure is that which exerts a bursting pressure on the walls of the containing vessel, and is the amount necessary to overcome the resistance to the flow of the fluid. The velocity pressure is that pressure which would be produced by the velocity if the air were brought to rest without shock or impact and which has to be used to produce this velocity. The total pressure is the sum of the two, i.e., the static and the velocity pressures. The velocity pressure, coming from the velocity and the density of the air, is given for 70 deg. F. by the following and from Fig. 329.

TABLE 102

| Velocity in Feet per Minute | Velocity Head |
|---|---|
| 750 | 0.004 |
| 1200 | 0.09 |
| 1500 | 0.14 |
| 1800 | 0.21 |
| 2100 | 0.28 |
| 2400 | 0.37 |
| 2700 | 0.46 |
| 3000 | 0.57 |

In designing a duct system there are two general methods of procedure: to fix on the velocity of the air in the different parts of the system taken from standard practice, and to find the total pressure drop by adding up the resistances to the flow in the different sections or to select a suitable friction pressure drop near the outlet and keep this constant by selecting the duct sizes in the separate sections on the basis of this constant pressure drop and the required quantity of air flowing. The second method is the most convenient and the quickest, and its application is shown by the following:

Referring to Fig. 327 it will be seen that 18 outlets are to be used. The amount of air per outlet is 22,000 ÷ 18 = 1220 cu. ft. per minute and the velocity through a 14-in. outlet is 1135 ft. per minute. Charts for air flow are usually made out for 100 ft. of straight pipe or duct and for ideally perfect construction and freedom from flaws. To allow for rough edges, or other faulty construction, a factor is usually added, frequently 50 per cent. In addition, changes in the cross-section or the direction of flow increase the resistance and this increase is usually given in the percentage of the velocity head. Variations such as are usual are given in Fig. 331. Using the chart and the figure of the duct layout, Table 103 is obtained.

TABLE 103

| Section | Volume of Air in Section, Cubic Feet per Minute | Diameter of Duct in Inches | Velocity of Air in Feet per Minute | Length of Section and Allowance for Ells, Etc. |
|---|---|---|---|---|
| a | 2,440 | 17 | 1550 | 22 ft. + 0.09 × 0.17 in. of water |
| b | 4,860 | 22¼ | 1790 | 22 ft. |
| c | 7,290 | 26¼ | 1920 | 22 ft. |
| d | 9,720 | 30 | 1980 | 22 ft. + 0.15 × 0.23 in. of water. |
| e | 22,000 | 42 | 2390 | 44 ft. |
| Total | ..... | ..... | ..... | 132 ft. + 0.049 in. of water. |

Taking a uniform resistance to air flow of 0.14 in. of water per 100 ft. of duct, the total resistance will be (132 ÷ 100 × 0.14) + 0.049 = 0.234 in. of water, and with an allowance for roughness, poor workmanship, etc., of 50 per cent. the resistance will be 0.234 × 1.5 = 0.351 in. of water for the duct only. In addition, there will be the resistance to the flow in the spray chamber of 0.25 in., and there will be the residual velocity of the air leaving the duct which will require, for 1140 ft. per minute velocity of the air in the branch, a static pressure of 0.090 in. of water, making a total pressure on the fan of 0.691. In designing a duct system it is usual to provide for the return as well as the delivery of the air. Quite frequently the return duct will be practically a duplicate of the delivery duct, in which there will be an additional friction to overcome and a correspondingly greater pressure. In the present problem, as the entire plant is located in the same room, it does not appear to be necessary to put in the return duct. The design shown will work satisfactorily.

In the present problem the temperature of the air is 51 deg. F. in the duct, and the calculations for the total pressure must be increased by an amount sufficient to allow for the increased density over that at the standard temperature of 70 deg. F., which is 1.021 in this case. Using this multiplying factor the total pressure head becomes 0.691 × 1.021 = 0.706 in. of water. The static pressure at the fan is some percentage of the total pressure, and taking the ratio of the total to the static pressure as 1.26 the static pressure becomes 0.706 ÷ 1.26 = 0.560. A fan selected for this static pressure will be chosen as well as a capacity of 22,000 cu. ft.

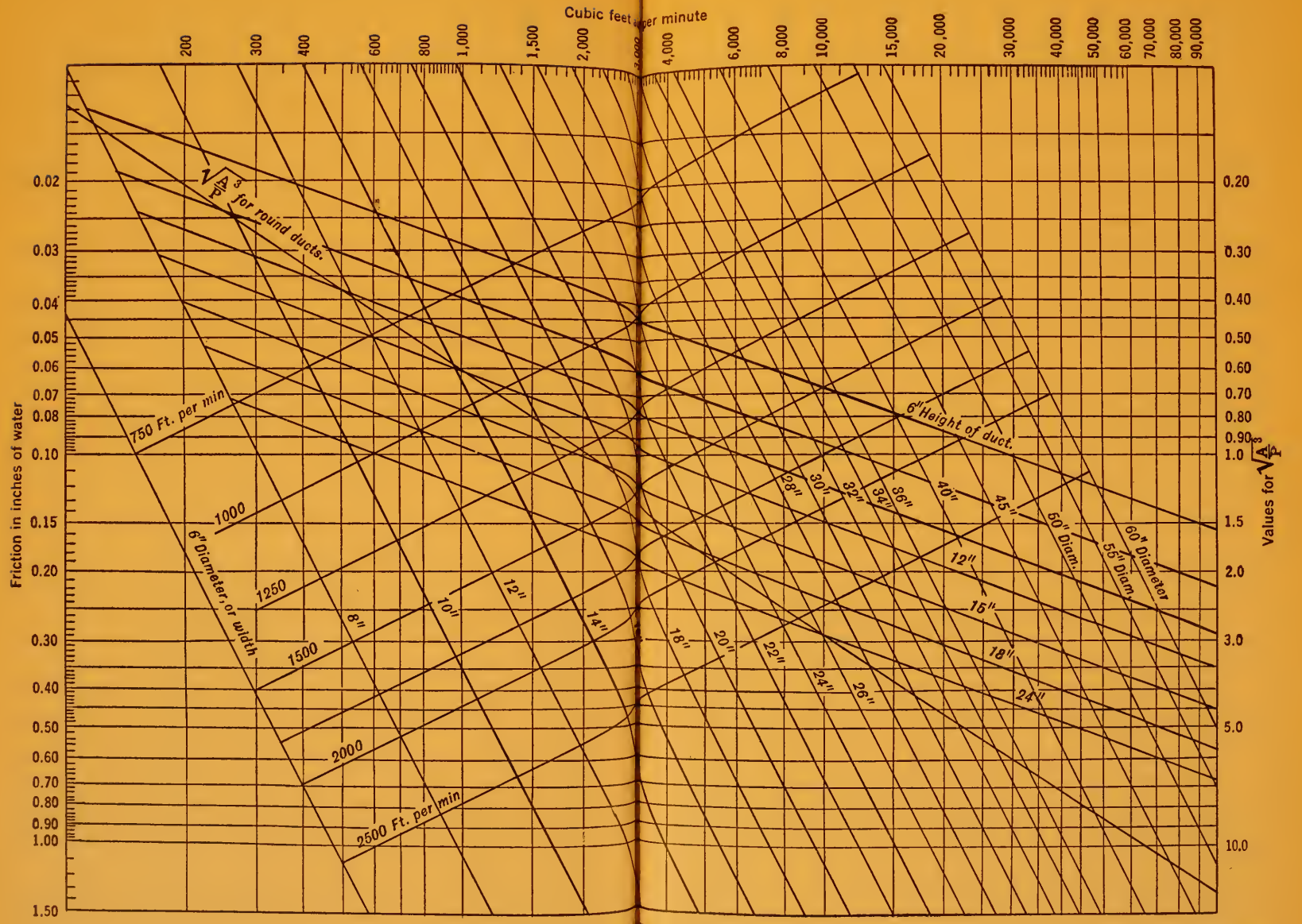



FIG. 330.—Drop of Pressure Due to Flow of Air in Ducts in Inches of Water.

*To face page 521.*

per minute. It is well to note that the tabulated values for fan performance are given at 70 deg. F. For operation at other temperatures the following table will be of value:

TABLE 104

| Temperature of the Air | Dividing Factor | Multiplying Factor |
|---|---|---|
| 60 | 0.99 | 1.010 |
| 50 | 0.98 | 1.021 |
| 40 | 0.97 | 1.032 |
| 30 | 0.96 | 1.042 |
| 20 | 0.95 | 1.053 |

It will be noticed that the velocities shown in the calculation increased as the size of the duct increased; from a little more than 1600 ft. at 17 in. diameter to a little less than 2400 ft. per minute at 42 in. diameter of duct. Also that the 30-degree branch outlet resulted in the same loss of head as would have been experienced with nearly 24 ft. of 17 in. diameter duct. Too much emphasis cannot be given to the desirability of making the number of changes in the cross-section of the duct a minimum. Right-angled turns should be avoided at all times, and when the size of the duct is to be changed it should be done gradually.

In the problem a round duct was chosen. Should it be desired to use a rectangular one with the same resistance per foot of length the same chart as for the round ducts can be used, making use of the relation $Q = .667h^{\frac{1}{2}}\left(\frac{A^3}{P}\right)^{\frac{1}{2}}$. For example, Fig. 330 has a line labeled $\left(\frac{A^3}{P}\right)^{\frac{1}{2}}$. To illustrate the use of the chart, suppose it is required to find the height of a rectangular duct if the width of the $b$ section is taken as 30 in. The quantity of air flowing in this section is 4880 cu. ft. per minute. Take the same resistance as before, 0.14 in. per 100 ft., and continue upwards on the chart until the quantity 4880 is reached. Then follow parallel with the diameter lines until the line $\left(\frac{A^3}{P}\right)^{\frac{1}{2}}$ is reached and then vertically upwards until the width of duct of 30 in. is reached. This point gives the desired answer of 16 in. for the height of the duct.

**Air Cooling in Blast Furnaces.**—Some three or four installations of the "Gayley Dry Air Blast" process were made in the United States and one in Wales previous to 1910. The Gayley process was an attempt to increase the economy of the furnace and was successful in certain respects, but the process has not been continued because other economies in other directions were found to be as good but not so expensive in first and operating costs. However, as an example of a method to be applied to refrigeration the following will be of interest.

The Gayley dry-blast process was developed about 1900. It consisted in cooling the air to a dew point temperature of 25 deg. F. at atmospheric pressure at which time the moisture content would be 19.5 grains per pound of dry air. The claims made for the process are that the furnace can work with greater regularity, and that a reduction of $12\frac{1}{2}$ per cent of the fuel required and a gain in 10 per cent in the production of the furnace is possible by its use. The reason advanced for this phenomenal gain in economy is that the moisture in the air (which at times may be 150 grains per pound of dry air) chills off the fire, just when the furnace temperature should be the hottest, by the chemical action of breaking up the water vapor into oxygen and hydrogen. The result

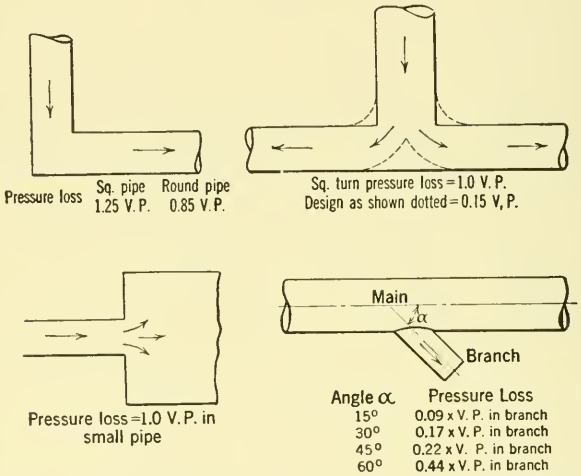

Pressure loss
Sq. pipe 1.25 V.P.
Round pipe 0.85 V.P.
Sq. turn pressure loss = 1.0 V.P.
Design as shown dotted = 0.15 V.P.
Pressure loss = 1.0 V.P. in small pipe
Main
α
Branch
Angle α
Pressure Loss
15° 0.09 x V.P. in branch
30° 0.17 x V.P. in branch
45° 0.22 x V.P. in branch
60° 0.44 x V.P. in branch

Fig. 331.—Loss of Velocity Head Due to Change of Section or Direction of Flow.

of this endothermic action is that there is not the proper removal of the sulphur or the proper deoxidation of the silicon during warm damp weather unless there is additional heat provided in the furnace by the addition of extra coke.

In the blast furnace it is necessary to use large blowing engines in order to compress the air required for supporting the combustion action in the furnace, and the blast may be at some pressure between 15 and 30 lb. gage. The compressed air is made to pass through checker work, previously heated by the exhaust gases from the furnace, until the blast temperature has become nearly 3000 deg. F. The refrigeration has been supplied as a rule on the suction side of the blowing engine, and therefore under these circumstances it will take place at approximately atmospheric pressure. The air cooled to 25 deg. F., with a uniform

water content of 19.5 grains per pound of dry air, is then passed to the compressor which is reduced in size thereby, on account of the increased density of the air.

The manner of cooling the air at the Isabella Works of the Carnegie Steel Company, but not operated since 1912, was to have two large bunker rooms with ammonia direct expansion piping in each, but so arranged as to permit the air to pass through one and then the other of these rooms by the operation of large butterfly valves, the reversal of the direction of flow thereby permitting the frost accumulation on the pipes to melt off in a manner similar to the present-time dehydrator of high pressure air for can ice agitation. Since 1912 the design of air cooling has changed decidedly so as to use the principle of the cooling tower or scrubber, arranged with tile wetted with water in the first stage and by brine in the second stage. Later came the design using sprays. This latter process, used in the Northern Steel Company, is shown in Fig. 332. In this case the air is compressed by the blowing tubs first, and in consequence of the higher pressure the air does not have to be cooled as low in temperature to secure the same weight of water content. For example, by compressing the air to 20 lb. gage it has been found necessary to cool the air down only to 46 deg. F. in order to secure the same weight of moisture as would be secured by cooling the air at atmospheric pressure down to 25 deg. F. In this case water can be used in the spray nozzles for the entire cooling action, and there will not be any necessity for reconcentration as would be required in the case of brine. Refrigeration is more economical because of the higher suction pressure being carried.

The principle underlying the liquefaction under pressure is as follows: Take a condition of air, saturated with water vapor, at 41 deg. F., at one atmosphere.

| | |
|---|---|
| Pressure of water vapor (at 41 deg. F.) | = 0.127 lb. |
| Pressure of the air = 14.69 − 0.127 | = 14.56 lb. |
| Grains of moisture per 1 lb. of dry air | = 37.87 |
| Volume of 1 lb. dry air | = 12.62 cu. ft. |

Now let the temperature remain constant at 41 deg. F. and compress the mixture up to 10 lb. gage or 24.69 lb. absolute, the air remaining saturated with moisture all the time. Then 24.69 − 0.13 = 24.56 lb. is the new pressure of the air content of the mixture.

$$\text{The new air volume} = 12.62 \times \frac{14.56}{24.56} = 7.49 \text{ cu. ft.}$$

The condensation of the water vapor occurs in the same ratio or

$$37.87 \times \frac{14.56}{24.56} = 22.45 \text{ grains.}$$

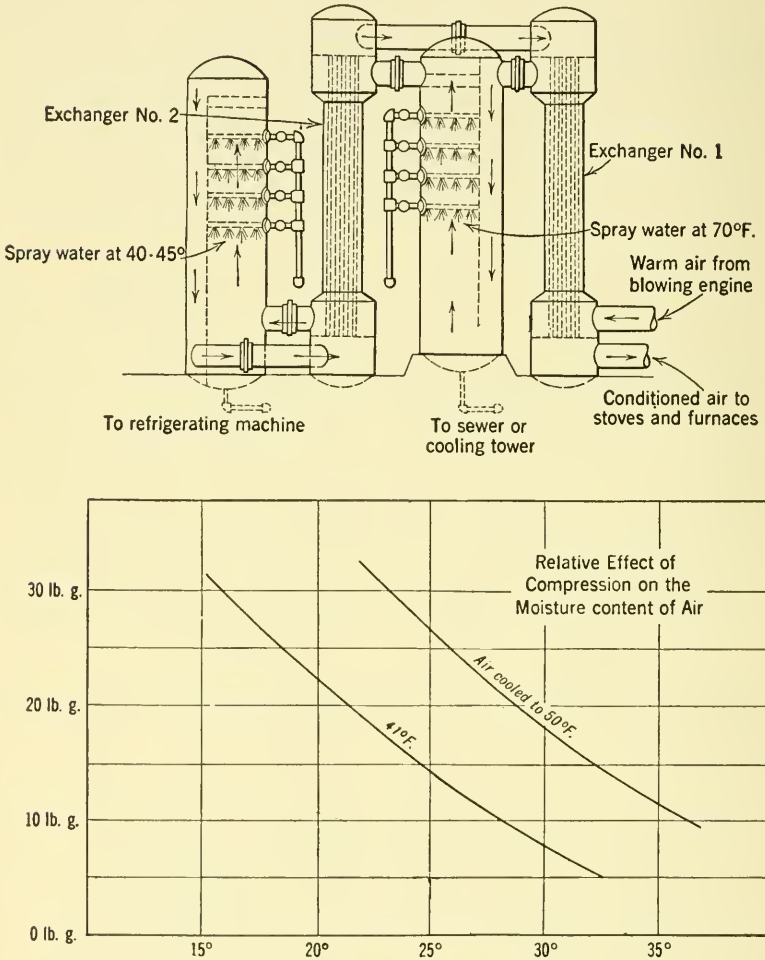



FIG. 332.—Air Cooling under Pressure.

However, this moisture content, 22.45 grains, is the amount which would give saturation with a pressure of one atmosphere air pressure and a temperature of a little more than 28 deg. F. Figure 332 gives an idea of the effect of compression and moderate cooling on the water vapor in the air. The principle just outlined occurs also in the de-

humidifier in the manufacture of ice with high-pressure air agitation, and in the synthetic process for the manufacture of ammonia.

**The Cooling of Theatres and Public Buildings.**—Mechanical refrigeration has been able to provide a means of cooling the air so that the public dining room, banquet hall and theatre are reasonably comfortable and suitable for service during the hottest part of the summer. The same may be said also in regard to banks, offices and such places of congested gathering as the New York Stock Exchange, etc. In none of these places is there a desire to chill the air very much, but simply to make it feel cool on entering. Therefore some such plan as the following would be used: The temperature of the chilled air will be 80 degrees when the outside air is 90, 75 when the outside air is 85, and 72 when the outside air temperature is 80 deg. F., and the relative humidity will not be greater than 60 per cent when the temperature in the theatre is 80 degrees and 65 per cent when the air is 72 degrees. In none of these cases is the air cooled more than 10 deg. F., and the result is a feeling of comfort on entering the building, but there must never be a sensation of chill. The procedure in such cases of cooling is to use the same equipment for ventilation and cooling as for heating and ventilation, the air being washed and a certain percentage, 50 to 67 per cent, being circulated. Such cooling plants are of large capacities, requiring as they do about $2\frac{1}{2}$ tons of refrigeration per 1000 cu. ft. per minute of fan capacity for localities similar to the summer conditions found in Chicago.

*The Refrigerating Machine.*—The refrigerating machine is now (1928) always a compressor, and the refrigerant has nearly always been carbon dioxide because of the lack of danger due to its use.[4] As the air in the spray chamber is seldom, if ever, cooled below 55 deg. F. (using 47-degree water in the sprays), the refrigerating machine can operate at the high boiling temperature of 30 to 32 deg. F., and so in consequence of which the capacity of the compressor is high as compared with the standard conditions of 5 deg. F. and the horse power per ton of refrigeration is reduced. For some time the method of cooling has been to install direct expansion piping at the entrance of the spray chamber using an amount equal to 35 lin. ft. of $1\frac{1}{4}$-in. pipe per ton of refrigeration. The piping is used both to cool the entering air and also to cool the water which is showered over the coils and which act as a "Baudalot" cooler. With such an arrangement it would be out of the question to use ammonia because of the possible panic should some accident occur to these coils and ammonia gas enter the ventilating ducts.

[4] Tests reported to have been performed by the Board of Trade on the SS. *Braemar Castle* indicated that no discomfort was experienced when the entire charge of carbon dioxide was released in the engine room, nor were lighted candles affected in any manner.

The air leaving the spray chamber will have an assumed temperature of 55 degrees and 100 per cent relative humidity, but during the passage through the fan and the distributing ducts the temperature will rise about 8 degrees, depending on the friction losses and heat leakage,

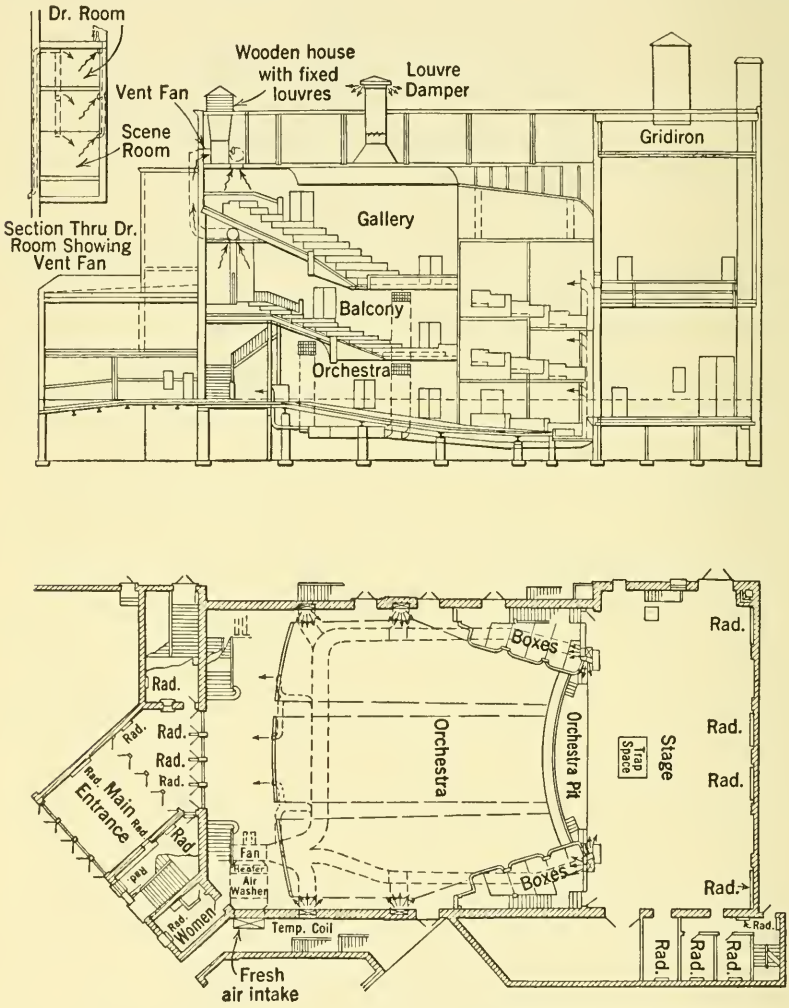



Fig. 333.—Auditorium Cooling and Ventilation.

which will give the air a dry bulb temperature of 63 degrees and a relative humidity of 75 per cent at the point of entrance to the hall or the auditorium. The air may be admitted to the auditorium either by the use of " mushrooms " in the floor or by the use of ceiling registers

and downward draft and in either case about 35 cu. ft. of air per minute per person is supplied. The fresh air from the sprays mixes with the air in the auditorium, and, if the conditions are such as the installation was designed for, the temperature will be 80 degrees and 60 per cent relative humidity. In the case of banquet halls and special dining rooms, the air can be designed to enter through registers in the walls, some 6 or 7 ft. from the floor, or from registers in the columns.

*The Spray Chamber.*—The air is cooled always by the use of water properly atomized by passing through nozzles especially designed for the purpose, usually with 15 to 25 lb. static pressure on the nozzle. Some engineers use the rule 8 gal. of water per 1000 cu. ft. of air per minute, while others use the rule of 1 lb. of water per 1 lb. of air. If the water is permitted to rise 8 deg. F. in temperature, 25 lb. or 8 gal. per minute would be required. A "Carrier" nozzle with a $\frac{3}{16}$-in. opening will deliver 15 lb. of water at a pressure of 20 lb. per sq. in., and 17 lb. at a pressure of 25 lb. per sq. in., whereas a $\frac{1}{4}$-in. opening will deliver 26 and 29 lb., respectively, at these pressures. From Table 101 it will be be seen that the Sturtevant and the Webster nozzles have about the same capacities. The details of the spray chamber are the same as in the previous problem.

**Problem.**—It is desired to install a cooling plant in a 3000-seat theatre for summer conditions of 90 deg. F. dry bulb and 60 per cent relative humidity outside conditions, 80 degree dry bulb inside and 63 degree dew point temperature. The heating effect of the illumination will be taken as 1500 B.t.u. per minute, and the heat leakage through the roof, walls, etc., will be taken as 3000 B.t.u. per minute. The sensible heat will be taken at 7.0 B.t.u. per minute per person, and the amount of water vapor emitted as 18.9 grains. This last is a high value but it is justified during hot weather. These heating values may be summed up as follows:

| | B.t.u. |
|---|---|
| *a*. The heating effect of the audience (the sensible heat) = 7.0 × 3000 | = 21,000 |
| *b*. The heat leakage (assumed) | = 3,000 |
| *c*. The illumination (assumed) | = 1,500 |
| *d*. The heat equivalent of the work of the fan (estimated at 45 hp.) | = 1,910 |
| Total refrigeration per minute | = 27,410 |

If the specific heat of air at 63.0 degrees, saturated with moisture, is taken as 0.2472, and the rise of temperature of the air is 17.0 deg. F., the heat capacity of the air will be 17.0 × 0.2472 = 4.20 B.t.u. The total amount of air to be circulated by the fan to absorb 27,410 B.t.u. per minute is therefore 6526 lb. The volume of 1 lb. of air saturated with moisture at 63.0 degrees is 13.43 cu. ft., so that the total capacity of the fan per minute is 6526 × 13.43 or 87,640 cu. ft., and the amount supplied to each person, if at 63.0 degrees, is 29.2 cu. ft. per minute.

*The Refrigeration Required.*—The amount of water vapor emitted is 18.9 × 3000 ÷ 7000 = 8.1 lb. per minute, and the refrigeration required to condense this amount will be 8.1 × 1055 = 8546 B.t.u., thereby making a total of 35,956 B.t.u.

per minute, which would have to be removed from the air if all of it were to be recirculated. As one half is to be recirculated and the other half is exhausted into the atmosphere there will be 17,980 B.t.u. per minute of refrigeration required for

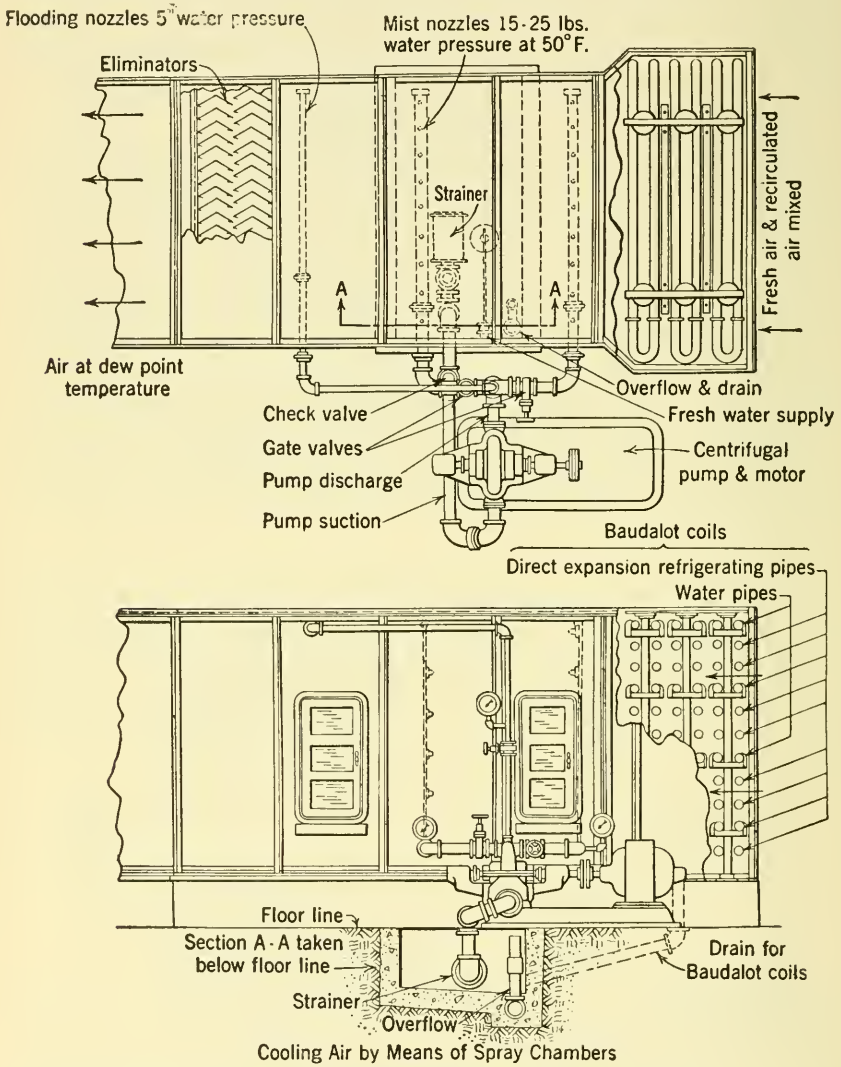



Fig. 334.—The Water Spray Air Cooling Chamber with Direct Expansion Coils for Carbon Dioxide Refrigeration.

the air to be recirculated. The fresh air will enter the spray chamber at 90 degrees and 60 per cent relative humidity, which corresponds to a dew point temperature of 74.2 and a moisture content of 127 grains per pound of dry air. The refrigeration

required to cool this air to the temperature of the spray chamber, taken from the total heat curve is: $40.9 - 28.0 = 12.9$ B.t.u. per pound, the total refrigeration is $6526 \div 2 \times 12.9 = 42{,}090$ B.t.u., and the total demand on the refrigeration equipment becomes $42{,}090 + 17{,}980 = 60{,}070$ B.t.u. Adding 5 per cent for safety, the amount becomes 63,070 B.t.u. per minute, or 315 tons of refrigeration.

It is clear from the calculation that the recirculated air, for the conditions stated in the problem, requires less cooling effect than does the fresh air from the outside, and that the amount of recirculation affects the load on the machine very decidedly. Therefore the operator has some reserve capacity in the machine available, on a pinch, by recirculating more air, if permitted by the city ordinances, as, for example, the circulation of 67 per cent instead of only 50 per cent.

If the spray water has a rise of temperature of 8 degrees there will be required 3 gal. per ton of refrigeration per minute, or $3.0 \times 315 = 945$ gal. total per

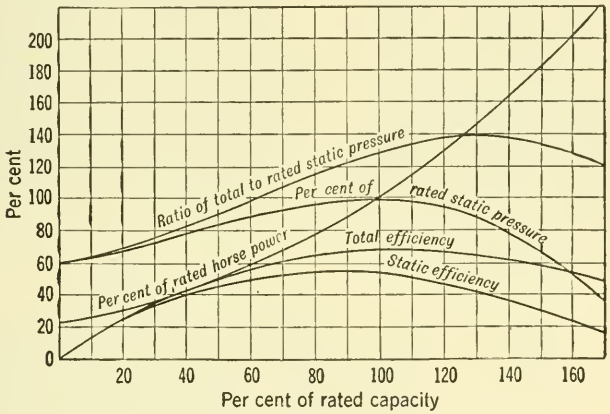



FIG. 335.—Characteristics of Conoidal Fans.

minute. Usually there is provided an additional amount, say 20 per cent, so that the pump will need to be 1134 gal. If a $\frac{1}{4}$-in. nozzle is used, which has a capacity of 29 lb. of water per minute with a pressure of 25 lb. per sq. in. at the nozzle, the number of sprays required will be $\frac{1134 \times 8.33}{29} = 326$. Using a velocity of 500 ft. per minute for the air in the spray chamber there will be required $87{,}640 \div 500 = 175.2$ sq. ft. of cross-sectional area, and if the spray chamber is limited to 10 ft. in height the width would need to be 17.5 ft. The arrangement of the nozzles, the amount of evaporating surface for cooling the water, the amount of eliminator surface, etc., will be found in a manner similar to that in the previous problem.

**Air Conditioning in Mines.**—According to Wilcox and Farmer,[5] the only mechanically refrigerated mine in the world is located in the Province of Minas Geraes, in Brazil. This mine has its pit head 2770 ft. above sea level and the mine depth in 1920 had been carried to 6400 ft.

[5] F. A. Wilcox and J. D. Farmer, Application of Refrigeration to the Ventilation of Mines, Fourth International Congress of Refrigeration.

where the rock temperature was recorded to be 118 deg. F. and the wet bulb temperature was from 90 to 95 deg. F., thereby making operating conditions very difficult for the workmen. Calculations showed that at a mine depth of 6500 ft., in order to insure not more than 81 degrees wet bulb temperature, the initial dew point temperature of the air would need to be 45.5 degrees, which would correspond to

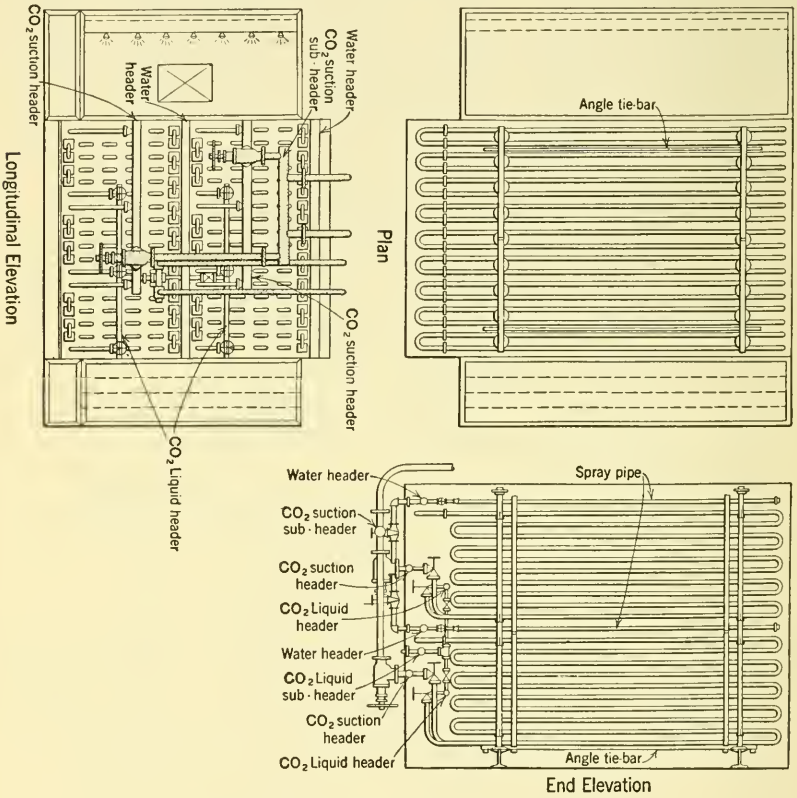



Fig. 336.—Air Cooling with Carbon Dioxide.

50 grains of water per one pound of dry air at 27.4 in. of the barometer. These calculations were based on a 34-in. barometer, a wet bulb temperature of 81 degrees, and a moisture content of 110 grains.

The plant selected was based on a fan capacity of 80,000 cu. ft. per minute and the air to be cooled to a dew point temperature of 43 deg. F., which represented a removal of 100,600 B.t.u. per minute, or 500 tons of refrigeration capacity. The load was arranged for with six units in order to permit economical operation at all seasons of the year, especially

at those times when the load would be less than the maximum. In this case water can be used satisfactorily for the cooling of the air, with a water temperature range of from 3 to 5 degrees. The method of cooling used in this installation was by the use of revolving cylinders whereby the air was brought into contact with a film of chilled water. The rock temperature, according to the mine records, was cooled from 118 to 107 deg. F.

**Carburetor Testing.**—Some manufacturers of automobiles have a cold temperature room in which a motor is operated under varying loads with various designs and adjustments of carburetors.[6] The cold temperature room is designed for − 10 deg. F. The automobile motor usually has the dynamometer outside the room and the cooling water for the radiator sometimes is piped in from the outside. The refrigeration required to maintain the temperature constant at − 10 degrees for a 60 hp. motor, may be calculated as follows, assuming an electrical dynamometer, 90 degrees outside, 6 in. of corkboard on the walls, ceiling and floors of a room 12 by 12 by 7 ft. Assume the walls are of 6-in. concrete. The heat balance of the motor may be taken as

| | Per cent |
|---|---|
| Engine friction | 4 |
| Radiation | 4 |
| Exhaust | 40 |
| Jacket | 27 |
| Useful work | 25 |

Even with the exhaust piping well lagged some heat will be lost to the room, and this will be taken as 25 per cent. Taking the efficiency of the electric dynamometer as 95 per cent, 5 per cent of the brake hp. will appear as heat in the electric windings. The amount of air required, to be brought in from the outside, will be taken as 15 lb. per pound of gasoline, and 0.7 pint of gasoline[7] will be required per b.hp. hour. The heat entering the room is then

$$Q = (0.04 + 0.04 + 0.25 \times 0.40)\left(\frac{60 \times 42.4}{0.25}\right) + (0.05 \times 60 \times 42.4) + \left(\frac{47.5 \times 30.7 \times 15}{60}\right) + \left(\frac{624 \times 0.05 \times 90 + 10}{60}\right)$$

$$= 2380 \text{ B.t.u. per minute} = 11.9 \text{ tons.}$$

[6] This problem was suggested by R. N. Cole, of Canton, Ohio.

[7] Take the specific gravity of gasoline as 0.702 and 1.0 pint of gasoline as weighing 0.732 lb.

Adding 10 per cent for safety, the total amount of refrigeration which should be provided will be 13.1 tons.

**Selection of Fan.**—The essential factors in the selection of a fan are: the capacity of the fan in cubic feet per minute; the static pressure developed at the fan discharge; and the horse power required to drive the fan. Each type of design has its own characteristic curves, and these are dependent on the curvature of the blade tip which may be radial, forward, backward and steel plate. The short curved blade is the

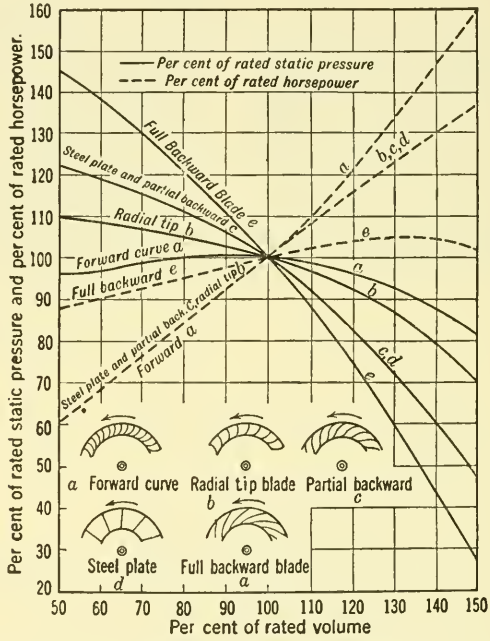



Fig. 337.—Characteristic Curves for Standard Fan Tips.

favorite with refrigerating engineers because of its ability to run at moderately high speeds.

The *steel plate* is for slow speeds only, and usually is steam engine driven. It is not an efficient type and is not used in refrigeration to any extent.

The *full forward* tip fan has a high outlet velocity but is noisy. It has the slowest tip speed for a given tip velocity of any design, but the power required increases rapidly with the speed, and it requires an oversized motor to prevent danger in the burning out of the motor windings.

The *full backward* tip fan has a rising horse power up to a certain

capacity and then drops off. The efficiency is high and this design is satisfactory in every respect for motor drives.

After the type of design of the fan is selected the size and the speed must be chosen. Previous calculations will have determined the total pressure, and the fan catalogue will show the ratio of the static to the total pressure, the static pressure can then be calculated and the problem reduced to the selection of a fan for a known capacity and static pressure at discharge. Reference to the tables will seldom give the exact conditions required, but other conditions may be found from the following:

*a.* The blade tip velocity and the volume of air delivered varies *directly* as the revolutions per minute.

*b.* The static pressure developed varies as the *square* of the revolutions per minute.

*c.* The power required to drive the fan varies as the *cube* of the revolutions per minute.

It is quite evident that increasing the speed in order to get the required volume of air should be limited to small amounts unless the increased static pressure is also desired. For example, the Niagara No. 6 conoidal fan at 472 r.p.m. delivers a volume of 13,450 cu. ft. per minute and requires 5.8 hp., but No. 8 at 204 r.p.m. delivers 13,820 cu. ft. and requires only 1.98 hp.

TABLE 105

CAPACITIES OF BUFFALO NIAGARA CONOIDAL FANS (TYPE N) UNDER AVERAGE WORKING CONDITIONS

70 Deg. F. and 29.92-In. Barometer

| Size | Diameter of Blast Wheel, Inches | Area of Outlet, Square Feet | ½-In. Static Pressure =0.288 Oz. | | | ¾-In. Static Pressure =0.433 Oz. | | | 1-In. Static Pressure =0.577 Oz. | | | 1½-In. Static Pressure =0.865 Oz. | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | R.p.m. | Volume of cubic feet per minute | Horsepower | R.p.m. | Volume of cubic feet per minute | Horsepower | R.p.m. | Volume of cubic feet per minute | Horsepower | R.p.m. | Volume of cubic feet per minute | Horsepower |
| 3 | 15⅝ | 1.31 | 544 | 1,945 | 0.28 | 668 | 2,380 | 0.51 | 770 | 2,750 | 0.78 | 943 | 3,365 | 1.45 |
| 3½ | 18⅛ | 1.79 | 465 | 2,642 | 0.38 | 572 | 3,240 | 0.69 | 660 | 3,740 | 1.06 | 809 | 4,580 | 1.97 |
| 4 | 20½ | 2.33 | 408 | 3,459 | 0.50 | 500 | 4,230 | 0.90 | 577 | 4,895 | 1.39 | 709 | 5,980 | 2.58 |
| 4½ | 23½ | 2.95 | 362 | 4,375 | 0.63 | 445 | 5,350 | 1.14 | 514 | 6,195 | 1.75 | 630 | 7,575 | 3.26 |
| 5 | 26⅛ | 3.64 | 326 | 5,400 | 0.77 | 400 | 6,610 | 1.41 | 462 | 7,645 | 2.16 | 566 | 9,350 | 4.03 |
| 5½ | 28¾ | 4.41 | 296 | 6,540 | 0.94 | 364 | 8,000 | 1.71 | 420 | 9,250 | 2.62 | 515 | 11,320 | 4.87 |
| 6 | 31⅜ | 5.25 | 272 | 7,780 | 1.11 | 334 | 9,525 | 2.03 | 386 | 11,000 | 3.12 | 472 | 13,450 | 5.80 |
| 7 | 36½ | 7.14 | 233 | 10,590 | 1.52 | 286 | 12,950 | 2.77 | 330 | 14,980 | 4.24 | 405 | 18,330 | 7.90 |
| 8 | 42 | 9.33 | 204 | 13,820 | 1.98 | 250 | 16,910 | 3.61 | 289 | 19,550 | 5.54 | 354 | 23,950 | 10.30 |
| 9 | 47 | 11.81 | 181 | 17,500 | 2.51 | 222 | 21,400 | 4.57 | 256 | 24,750 | 7.01 | 314 | 30,300 | 13.05 |
| 10 | 52 | 14.58 | 103 | 21,600 | 3.09 | 200 | 26,450 | 5.65 | 231 | 30,550 | 8.65 | 283 | 37,400 | 16.10 |
| 11 | 58 | 17.64 | 148 | 26,150 | 3.74 | 182 | 32,000 | 6.85 | 210 | 37,000 | 10.48 | 257 | 45.250 | 19.48 |
| 12 | 63 | 21.00 | 136 | 31,100 | 4.45 | 167 | 38,100 | 8.15 | 193 | 44,050 | 12.48 | 236 | 53,900 | 23.20 |
| 13 | 68 | 24.65 | 125 | 36,500 | 5.22 | 154 | 44,700 | 9.56 | 178 | 51,650 | 14.62 | 217 | 63,200 | 27.20 |
| 14 | 73 | 28.68 | 116 | 42,350 | 6.06 | 143 | 51,900 | 11.08 | 165 | 60,000 | 16.96 | 202 | 73,200 | 31.55 |
| 15 | 78 | 32.80 | 109 | 48,550 | 6.95 | 133 | 59,500 | 12.70 | 154 | 68,850 | 19.49 | 189 | 84,100 | 36.25 |
| 16 | 84 | 37.32 | 102 | 55,300 | 7.91 | 125 | 67,750 | 14.46 | 144 | 78,300 | 22.15 | 177 | 95,750 | 41.20 |
| 17 | 89 | 42.14 | 96 | 62,500 | 8.95 | 118 | 76,500 | 16.32 | 136 | 88,400 | 25.00 | 167 | 108,000 | 46.50 |
| 18 | 94 | 42.24 | 91 | 70,000 | 10.01 | 111 | 85,600 | 18.30 | 128 | 99,100 | 28.05 | 157 | 121,200 | 52.15 |
| 19 | 99 | 52.63 | 86 | 78,000 | 11.15 | 105 | 95,500 | 20.40 | 122 | 110,200 | 31.25 | 149 | 135,000 | 58.05 |
| 20 | 105 | 58.32 | 82 | 86,450 | 12.36 | 100 | 105,850 | 22.60 | 116 | 122,200 | 34.65 | 142 | 149,500 | 64.45 |

TABLE 105—*Continued*

| Size | Diameter of Blast Wheel, Inches | Area of Outlet, Square Feet | 2-In. Static Pressure = 1.154 Oz. | | | 2½-In. Static Pressure = 1.442 Oz. | | | 3-In. Static Pressure = 1.734 Oz. | | | 3½-In. Static Pressure = 2.019 Oz. | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | R.p.m. | Volume of cubic feet per minute | Horsepower | R.p.m. | Volume of cubic feet per minute | Horsepower | R.p.m. | Volume of cubic feet per minute | Horsepower | R.p.m. | Volume of cubic feet per minute | Horsepower |
| 3 | 15⅝ | 1.31 | 1088 | 3,890 | 2.21 | 1215 | 4,350 | 3.08 | 1332 | 4,770 | 4.05 | 1443 | 5,150 | 5.13 |
| 3½ | 18⅛ | 1.79 | 934 | 5,300 | 3.01 | 1010 | 5,930 | 4.19 | 1141 | 6,495 | 5.53 | 1238 | 7,010 | 6.98 |
| 4 | 20½ | 2.33 | 817 | 6,920 | 3.93 | 912 | 7,730 | 5.47 | 1000 | 8,480 | 7.22 | 1082 | 9,160 | 9.12 |
| 4½ | 23½ | 2.95 | 726 | 8,750 | 4.97 | 810 | 9,795 | 6.93 | 890 | 10,740 | 9.14 | 964 | 11,590 | 11.55 |
| 5 | 26⅛ | 3.64 | 655 | 10,820 | 6.15 | 730 | 12,070 | 8.55 | 800 | 13,250 | 11.26 | 868 | 14,300 | 14.25 |
| 5½ | 28¾ | 4.41 | 595 | 13,100 | 7.43 | 664 | 14,600 | 10.35 | 728 | 16,030 | 13.62 | 789 | 17,300 | 17.25 |
| 6 | 31⅜ | 5.25 | 545 | 15,550 | 8.85 | 609 | 17,390 | 12.30 | 667 | 19,090 | 16.22 | 723 | 20,600 | 20.55 |
| 7 | 36½ | 7.14 | 468 | 21,200 | 12.02 | 522 | 23,650 | 16.75 | 572 | 26,000 | 22.10 | 620 | 28,050 | 27.95 |
| 8 | 42 | 9.33 | 409 | 27,650 | 15.70 | 456 | 30,900 | 21.90 | 500 | 33,950 | 28.85 | 542 | 36,600 | 36.50 |
| 9 | 47 | 11.81 | 364 | 35,050 | 19.90 | 405 | 39,100 | 27.70 | 445 | 42,950 | 36.55 | 482 | 46,350 | 46.20 |
| 10 | 52 | 14.58 | 327 | 43,250 | 24.55 | 365 | 48,300 | 34.20 | 400 | 53,000 | 45.15 | 433 | 57,200 | 57.00 |
| 11 | 58 | 17.64 | 297 | 52,300 | 29.70 | 332 | 58,450 | 41.45 | 364 | 64,100 | 54.60 | 394 | 69,300 | 69.00 |
| 12 | 63 | 21.00 | 272 | 62,300 | 35.50 | 304 | 69,550 | 49.25 | 334 | 76,400 | 65.00 | 361 | 82,500 | 82.15 |
| 13 | 68 | 24.65 | 252 | 73,050 | 41.50 | 280 | 81,600 | 57.80 | 308 | 89,550 | 76.30 | 334 | 96,750 | 96.45 |
| 14 | 73 | 28.68 | 234 | 84,900 | 48.15 | 261 | 94,600 | 67.05 | 286 | 103,900 | 88.70 | 310 | 112,050 | 111.90 |
| 15 | 78 | 32.80 | 218 | 97,250 | 55.25 | 243 | 108,700 | 77.00 | 267 | 119,200 | 101.50 | 289 | 128,800 | 128.20 |
| 16 | 84 | 37.32 | 204 | 110,750 | 62.85 | 228 | 123,600 | 87.50 | 250 | 135,800 | 115.50 | 271 | 146,400 | 146.00 |
| 17 | 89 | 42.14 | 192 | 125,000 | 71.00 | 214 | 139,500 | 99.00 | 235 | 153,100 | 130.30 | 255 | 165,300 | 164.80 |
| 18 | 94 | 47.24 | 182 | 140,000 | 79.50 | 203 | 156,500 | 110.80 | 222 | 171,800 | 146.00 | 241 | 185,300 | 184.60 |
| 19 | 99 | 52.63 | 172 | 156,000 | 88.55 | 192 | 174,200 | 123.40 | 211 | 191,200 | 162.80 | 228 | 206,200 | 206.00 |
| 20 | 105 | 58.32 | 164 | 173,000 | 98.25 | 183 | 193,000 | 136.80 | 200 | 212,000 | 180.30 | 217 | 229,000 | 228.00 |

Total Pressure is 127.4 per cent of the Rated Static Pressure.

TABLE 105*a*

CAPACITIES OF BUFFALO TURBO CONOIDAL FANS (TYPE T) UNDER AVERAGE WORKING CONDITIONS

70 Deg. F. and 29.92-In. Barometer

| Size | Diameter of Blast Wheel, Inches | Area of Outlet, Square Feet | ½-In. Static Pressure =0.288 Oz. | | | ¾-In. Static Pressure =0.433 Oz. | | | 1-In. Static Pressure =0.577 Oz. | | | 1½-In. Static Pressure =0.865 Oz. | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | R.p.m. | Volume of cubic feet per minute | Horsepower | R.p.m. | Volume of cubic feet per minute | Horsepower | R.p.m. | Volume of cubic feet per minute | Horsepower | R.p.m. | Volume of cubic feet per minute | Horsepower |
| 2½ | 14¼ | 0.91 | 1115 | 1,230 | 0.20 | 1368 | 1,500 | 0.36 | 1580 | 1,740 | 0.56 | 1935 | 2,120 | 1.03 |
| 3 | 17¼ | 1.31 | 930 | 1,770 | 0.28 | 1140 | 2,160 | 0.52 | 1315 | 2,500 | 0.81 | 1610 | 3,060 | 1.48 |
| 3½ | 20 | 1.79 | 797 | 2,410 | 0.39 | 976 | 2,940 | 0.71 | 1130 | 3,410 | 1.10 | 1380 | 4,160 | 2.02 |
| 4 | 22¾ | 2.33 | 697 | 3,140 | 0.51 | 855 | 3,850 | 0.93 | 987 | 4,450 | 1.44 | 1208 | 5,440 | 2.64 |
| 4½ | 25¾ | 2.95 | 620 | 3,980 | 0.64 | 760 | 4,860 | 1.18 | 879 | 5,640 | 1.82 | 1075 | 6,890 | 3.34 |
| 5 | 28½ | 3.64 | 558 | 4,910 | 0.79 | 684 | 6,000 | 1.45 | 790 | 6,950 | 2.25 | 966 | 8,500 | 4.12 |
| 5½ | 31½ | 4.41 | 507 | 5,950 | 0.96 | 621 | 7,270 | 1.76 | 719 | 8,400 | 2.72 | 880 | 10,300 | 5.00 |
| 6 | 34¼ | 5.25 | 465 | 7,070 | 1.14 | 570 | 8,650 | 2.09 | 658 | 10,000 | 3.24 | 806 | 12,230 | 5.77 |
| 6½ | 36¾ | 6.16 | 430 | 8,300 | 1.33 | 526 | 10,200 | 2.46 | 608 | 11,750 | 3.80 | 743 | 14,350 | 6.96 |
| 7 | 39¾ | 7.14 | 398 | 9,630 | 1.55 | 488 | 11,780 | 2.85 | 565 | 13,610 | 5.40 | 690 | 16,650 | 8.09 |
| 7½ | 42½ | 8.19 | 372 | 11,050 | 1.78 | 456 | 13,500 | 3.27 | 526 | 15,610 | 5.05 | 645 | 19,100 | 9.27 |
| 8 | 45½ | 9.33 | 349 | 12,590 | 2.02 | 428 | 15,370 | 3.72 | 495 | 17,800 | 5.75 | 604 | 21,750 | 10.55 |
| 8½ | 48 | 10.53 | 328 | 14,200 | 2.28 | 402 | 17,380 | 4.21 | 465 | 20,100 | 6.50 | 569 | 24,600 | 11.90 |
| 9 | 51¼ | 11.81 | 310 | 15,900 | 2.56 | 380 | 19,450 | 4.71 | 440 | 22,500 | 7.29 | 536 | 27,500 | 13.35 |
| 10 | 56¾ | 14.58 | 279 | 19,650 | 3.16 | 342 | 24,050 | 5.82 | 395 | 27,800 | 9.00 | 483 | 34,000 | 16.50 |
| 11 | 62½ | 17.64 | 254 | 23,800 | 3.82 | 311 | 29,100 | 7.05 | 359 | 33,700 | 10.90 | 439 | 41,100 | 19.95 |
| 12 | 68 | 21.00 | 232 | 28,300 | 4.55 | 286 | 34,600 | 8.40 | 329 | 40,100 | 12.95 | 402 | 49,000 | 23.80 |
| 13 | 73½ | 24.65 | 214 | 33,200 | 5.34 | 263 | 40,600 | 9.85 | 304 | 47,000 | 15.20 | 372 | 57,500 | 27.90 |
| 14 | 79 | 28.68 | 198 | 38,500 | 6.20 | 244 | 47,100 | 11.40 | 282 | 54,500 | 17.62 | 345 | 66,700 | 32.35 |
| 15 | 84¾ | 32.80 | 186 | 44,200 | 7.11 | 228 | 54,050 | 13.08 | 264 | 62,600 | 20.20 | 322 | 76,500 | 37.15 |
| 16 | 90¼ | 37.32 | 174 | 50,300 | 8.09 | 214 | 61,500 | 14.90 | 247 | 71,200 | 23.00 | 302 | 87,100 | 42.25 |

TABLE 105*a*—*Continued*

| Size | Diameter of Blast Wheel, Inches | Area of Outlet, Square Feet | 2-In. Static Pressure = 1.154 Oz. | | | 2½-In. Static Pressure = 1.442 Oz. | | | 3-In. Static Pressure = 1.734 Oz. | | | 3½-In. Static Pressure = 2.019 Oz. | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | R.p.m. | Volume of cubic feet per minute | Horsepower | R.p.m. | Volume of cubic feet per minute | Horsepower | R.p.m. | Volume of cubic feet per minute | Horsepower | R.p.m. | Volume of cubic feet per minute | Horsepower |
| 2½ | 14¼ | 0.91 | 2225 | 2,455 | 1.59 | 2490 | 2,750 | 2.22 | 2740 | 3,010 | 2.94 | 2958 | 3,250 | 3.69 |
| 3 | 17¼ | 1.31 | 1860 | 3,540 | 2.29 | 2075 | 3,960 | 3.19 | 2282 | 4,330 | 4.23 | 2463 | 4,680 | 5.30 |
| 3½ | 20 | 1.79 | 1595 | 4,800 | 3.12 | 1780 | 5,390 | 4.35 | 1958 | 5,890 | 5.75 | 2115 | 6,360 | 7.22 |
| 4 | 22¾ | 2.33 | 1395 | 6,270 | 4.08 | 1559 | 7,050 | 5.68 | 1713 | 7,700 | 7.52 | 1850 | 8,320 | 9.45 |
| 4½ | 25¾ | 2.95 | 1240 | 7,950 | 5.16 | 1385 | 8,920 | 7.19 | 1522 | 9,740 | 9.52 | 1645 | 10,550 | 11.95 |
| 5 | 28½ | 3.64 | 1117 | 9,800 | 6.37 | 1249 | 11,000 | 8.87 | 1370 | 12,000 | 11.75 | 1480 | 13,000 | 14.75 |
| 5½ | 31½ | 4.41 | 1015 | 11,880 | 7.72 | 1133 | 13,300 | 10.75 | 1245 | 14,550 | 14.23 | 1345 | 15,750 | 17.85 |
| 6 | 34¼ | 5.25 | 932 | 14,120 | 9.18 | 1040 | 15,800 | 12.78 | 1141 | 17,300 | 16.92 | 1232 | 18,700 | 21.25 |
| 6½ | 36¾ | 6.16 | 860 | 16,600 | 10.76 | 960 | 18,600 | 15.00 | 1054 | 20,300 | 19.85 | 1139 | 22,950 | 24.90 |
| 7 | 39¾ | 7.14 | 799 | 19,250 | 12.50 | 891 | 21,550 | 17.40 | 978 | 23,550 | 23.05 | 1056 | 25,450 | 28.90 |
| 7½ | 42½ | 8.19 | 745 | 22,100 | 14.32 | 831 | 24,750 | 19.95 | 914 | 27,050 | 26.40 | 987 | 29,200 | 33.20 |
| 8 | 45½ | 9.33 | 700 | 25,100 | 16.30 | 780 | 28,150 | 22.70 | 856 | 30,800 | 30.10 | 925 | 33,300 | 37.75 |
| 8½ | 48 | 10.53 | 657 | 28,400 | 18.40 | 736 | 31,800 | 25.60 | 807 | 34,750 | 33.95 | 870 | 37,550 | 42.25 |
| 9 | 51¼ | 11.81 | 621 | 31,800 | 20.65 | 693 | 35,600 | 28.75 | 761 | 38,950 | 38.05 | 822 | 41,050 | 47.80 |
| 10 | 56¾ | 14.58 | 559 | 39,300 | 25.50 | 625 | 44,000 | 35.50 | 685 | 48,100 | 47.00 | 740 | 52,000 | 59.00 |
| 11 | 62½ | 17.64 | 507 | 47,450 | 30.85 | 567 | 53,250 | 42.95 | 623 | 58,150 | 56.90 | 673 | 62,900 | 71.45 |
| 12 | 68 | 21.00 | 465 | 56,500 | 36.75 | 520 | 63,500 | 51.10 | 570 | 69,250 | 67.70 | 616 | 74,950 | 85.00 |
| 13 | 73½ | 24.65 | 430 | 66,200 | 43.05 | 480 | 74,400 | 60.00 | 527 | 81,300 | 79.40 | 569 | 87,900 | 99.60 |
| 14 | 79 | 28.68 | 399 | 76,800 | 50.00 | 445 | 86,300 | 69.55 | 489 | 94,300 | 92.10 | 528 | 101,900 | 115.60 |
| 15 | 84¾ | 32.80 | 373 | 88,500 | 57.40 | 415 | 99,000 | 80.00 | 456 | 108,000 | 105.70 | 493 | 117,000 | 133.00 |
| 16 | 90¼ | 37.32 | 349 | 100,500 | 65.30 | 390 | 112,500 | 90.90 | 428 | 123,000 | 120.50 | 462 | 133,000 | 151.00 |

Total Pressure is 122.7 per cent of the Rated Static Pressure.

TABLE 106

CAPACITIES OF BUFFALO PLANOIDAL STEEL PLATE BLOWERS (TYPE L) UNDER AVERAGE WORKING CONDITIONS

70 Deg. F. and 29.92-In. Barometer

| Size | Diameter of Blast Wheel, Inches | Area of Outlet, Square Feet | ½-In. Static Pressure = 0.288 Oz. | | | ¾-In. Static Pressure = 0.433 Oz. | | | 1-In. Static Pressure = 0.577 Oz. | | | 1½-In. Static Pressure = 0.865 Oz. | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | R.p.m. | Volume of cubic feet per minute | Horsepower | R.p.m. | Volume of cubic feet per minute | Horsepower | R.p.m. | Volume of cubic feet per minute | Horsepower | R.p.m. | Volume of cubic feet per minute | Horsepower |
| 30 | 19¼ | 0.77 | 678 | 1,160 | 0.23 | 830 | 1,420 | 0.42 | 958 | 1,640 | 0.65 | 1174 | 2,010 | 1.19 |
| 35 | 22½ | 1.04 | 580 | 1,570 | 0.31 | 710 | 1,925 | 0.58 | 820 | 2,220 | 0.87 | 1005 | 2,720 | 1.63 |
| 40 | 25¾ | 1.36 | 508 | 2,065 | 0.41 | 623 | 2,530 | 0.75 | 719 | 2,920 | 1.16 | 880 | 3,580 | 2.13 |
| 45 | 29⅝ | 1.75 | 451 | 2,600 | 0.52 | 553 | 3,185 | 0.96 | 639 | 3,680 | 1.47 | 783 | 4,510 | 2.70 |
| 50 | 32⅛ | 2.16 | 407 | 3,220 | 0.64 | 498 | 3,940 | 1.18 | 575 | 4,550 | 1.82 | 705 | 5,580 | 3.35 |
| 55 | 35⅜ | 2.61 | 369 | 3,890 | 0.77 | 452 | 4,765 | 1.42 | 522 | 5,500 | 2.19 | 640 | 6,740 | 4.03 |
| 60 | 38½ | 3.13 | 339 | 4,630 | 0.92 | 415 | 5,675 | 1.70 | 479 | 6,550 | 2.61 | 587 | 8,030 | 4.80 |
| 70 | 45 | 4.26 | 290 | 6,320 | 1.25 | 355 | 7,730 | 2.31 | 410 | 8,930 | 3.55 | 502 | 10,920 | 6.52 |
| 80 | 51⅜ | 5.54 | 254 | 8,230 | 1.64 | 315 | 10,080 | 3.02 | 359 | 11,630 | 4.65 | 440 | 14,250 | 8.55 |
| 90 | 57⅞ | 7.10 | 226 | 10,410 | 2.08 | 276 | 12,750 | 3.82 | 319 | 14,730 | 5.88 | 391 | 18,050 | 10.80 |
| 100 | 64¼ | 8.75 | 203 | 12,880 | 2.56 | 248 | 15,750 | 4.71 | 287 | 18,200 | 7.25 | 352 | 22,300 | 13.32 |
| 110 | 70¾ | 10.57 | 185 | 15,550 | 3.10 | 226 | 19,100 | 5.71 | 261 | 22,000 | 8.78 | 320 | 26,950 | 16.12 |
| 120 | 77¼ | 13.00 | 169 | 18,530 | 3.69 | 207 | 22,700 | 6.78 | 239 | 26,200 | 10.44 | 293 | 32,080 | 19.18 |
| 130 | 83½ | 14.85 | 156 | 21,600 | 4.31 | 192 | 26,450 | 7.93 | 221 | 30,550 | 12.20 | 271 | 37,410 | 22.40 |
| 140 | 90 | 17.20 | 145 | 25,200 | 5.02 | 177 | 30,850 | 9.24 | 205 | 35,650 | 14.20 | 251 | 43,700 | 26.10 |
| 150 | 96½ | 19.70 | 135 | 28,950 | 5.76 | 165 | 35,400 | 10.60 | 191 | 40,900 | 16.30 | 234 | 50,150 | 29.95 |
| 160 | 103 | 22.40 | 127 | 32,800 | 6.57 | 154 | 40,200 | 12.10 | 179 | 46,450 | 18.60 | 219 | 56,900 | 34.15 |
| 170 | 109¼ | 25.40 | 120 | 37,150 | 7.42 | 146 | 45,500 | 13.65 | 169 | 52,550 | 21.00 | 207 | 64,400 | 38.60 |
| 180 | 115¾ | 28.50 | 112 | 41,700 | 8.31 | 138 | 51,100 | 15.25 | 159 | 59,000 | 23.50 | 195 | 72,250 | 43.15 |
| 190 | 122¼ | 31.70 | 107 | 46,300 | 9.26 | 131 | 56,700 | 17.05 | 151 | 65,500 | 26.20 | 185 | 80,250 | 48.10 |
| 200 | 128½ | 35.30 | 102 | 51,500 | 10.25 | 125 | 63,100 | 18.85 | 144 | 72,850 | 29.00 | 176 | 89,200 | 53.30 |

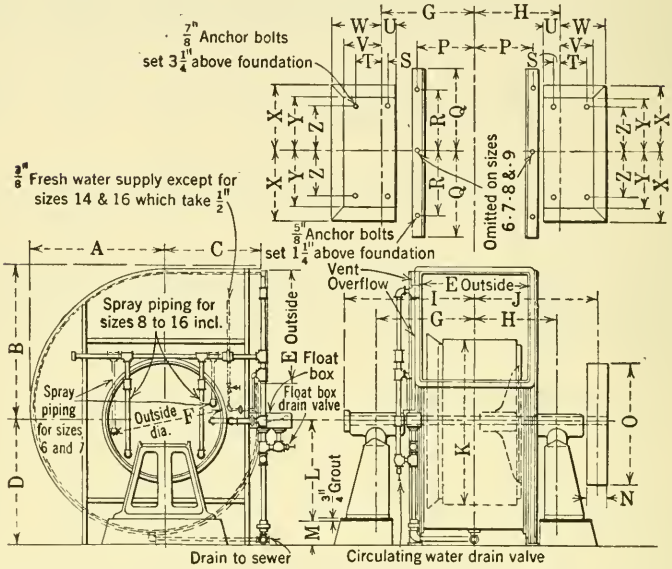
TABLE 106—*Continued*

| Size | Diameter of Blast Wheel, Inches | Area of Outlet, Square Feet | 2-In. Static Pressure = 1.154 Oz. | | | 2½-In. Static Pressure = 1.442 Oz. | | | 3-In. Static Pressure = 1.734 Oz. | | | 3½-In. Static Pressure = 2.019 Oz. | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | R.p.m. | Volume of cubic feet per minute | Horsepower | R.p.m. | Volume of cubic feet per minute | Horsepower | R.p.m. | Volume of cubic feet per minute | Horsepower | R.p.m. | Volume of cubic feet per minute | Horsepower |
| 30 | 19¼ | 0.77 | 1355 | 2,320 | 1.84 | 1515 | 2,595 | 2.57 | 1660 | 2,840 | 3.38 | 1792 | 3,070 | 4.26 |
| 35 | 22½ | 1.04 | 1160 | 3,140 | 2.52 | 1295 | 3,510 | 3.48 | 1420 | 3,845 | 4.63 | 1534 | 4,155 | 5.83 |
| 40 | 25¾ | 1.36 | 1018 | 4,135 | 3.28 | 1135 | 4,620 | 4.58 | 1245 | 5,060 | 6.03 | 1345 | 5,460 | 7.60 |
| 45 | 29⅞ | 1.75 | 904 | 5,210 | 4.15 | 1010 | 5,825 | 5.81 | 1108 | 6,375 | 7.63 | 1195 | 6,890 | 9.63 |
| 50 | 32⅛ | 2.16 | 814 | 6,440 | 5.15 | 910 | 7,200 | 7.20 | 996 | 7,880 | 9.45 | 1076 | 8,510 | 11.91 |
| 55 | 35⅜ | 2.61 | 738 | 7,780 | 6.19 | 826 | 8,700 | 8.66 | 904 | 9,530 | 11.38 | 976 | 10,290 | 14.34 |
| 60 | 38½ | 3.13 | 678 | 9,260 | 7.38 | 758 | 10,370 | 10.31 | 830 | 11,340 | 13.55 | 896 | 12,250 | 17.10 |
| 70 | 45 | 4.26 | 580 | 12,630 | 10.02 | 648 | 14,120 | 14.03 | 710 | 15,460 | 18.45 | 767 | 16,700 | 23.25 |
| 80 | 51⅜ | 5.54 | 508 | 16,450 | 13.12 | 568 | 18,400 | 18.40 | 621 | 20,150 | 24.20 | 672 | 21,750 | 30.50 |
| 90 | 57⅞ | 7.10 | 451 | 20,850 | 16.60 | 505 | 23,300 | 23.30 | 553 | 25,500 | 30.55 | 597 | 27,550 | 38.50 |
| 100 | 64¼ | 8.75 | 406 | 25,750 | 20.48 | 454 | 28,800 | 28.70 | 497 | 31,530 | 37.70 | 537 | 34,050 | 47.50 |
| 110 | 70¾ | 10.57 | 369 | 31,100 | 24.80 | 413 | 34,800 | 34.70 | 452 | 38,100 | 45.60 | 488 | 41,200 | 57.50 |
| 120 | 77¼ | 13.00 | 338 | 37,050 | 29.50 | 378 | 41,400 | 41.30 | 414 | 45,400 | 54.25 | 447 | 49,000 | 68.40 |
| 130 | 83½ | 14.85 | 313 | 43,250 | 34.50 | 350 | 48,350 | 48.25 | 383 | 52,900 | 63.40 | 413 | 57,200 | 80.00 |
| 140 | 90 | 17.20 | 290 | 50,400 | 40.15 | 324 | 56,400 | 56.15 | 355 | 61,750 | 73.80 | 384 | 66,700 | 93.00 |
| 150 | 96½ | 19.70 | 270 | 57,900 | 46.10 | 302 | 64,750 | 64.50 | 331 | 70,900 | 84.70 | 358 | 76,600 | 106.80 |
| 160 | 103 | 22.40 | 253 | 65,700 | 52.60 | 283 | 73,500 | 73.50 | 310 | 80,400 | 96.60 | 335 | 86,900 | 121.80 |
| 170 | 109¼ | 25.40 | 239 | 74,300 | 59.40 | 267 | 83,200 | 83.00 | 293 | 91,000 | 109.00 | 316 | 98,400 | 137.50 |
| 180 | 115¾ | 28.50 | 225 | 83,500 | 66.40 | 251 | 93,400 | 93.00 | 277 | 102,200 | 122.20 | 298 | 110,400 | 154.00 |
| 190 | 122¼ | 31.70 | 214 | 92,650 | 74.20 | 239 | 103,700 | 103.60 | 262 | 113,300 | 136.00 | 282 | 122,500 | 171.50 |
| 200 | 128½ | 35.30 | 204 | 103,000 | 82.00 | 228 | 115,100 | 114.70 | 250 | 126,100 | 150.80 | 269 | 136,300 | 190.00 |

Total Pressure is 126 per cent of the Rated Static Pressure.

## TABLE 107

### Dimensions of "ABC" Air Washing and Cooling Fans


7/8" Anchor bolts set 3 1/4" above foundation
3/8" Fresh water supply except for sizes 14 & 16 which take 1/2"
5/8" Anchor bolts set 1 1/4" above foundation
Omitted on sizes 6-7-8 & 9
Spray piping for sizes 8 to 16 incl.
Spray piping for sizes 6 and 7
Outside dia.
E Outside
E Float box
Float box drain valve
Vent
Overflow
3/4" Grout
Drain to sewer
Circulating water drain valve

| Fan Number | A | B | C | D | E | F | G | H | I | J | K | L | M | N | O |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 6 | 33 3/8 | 38 1/8 | 27 1/2 | 38 | 25 3/4 | 27 3/4 | 27 | 22 1/2 | 36 7/16 | 33 3/4 | 36 | 32 3/4 | 5 1/4 | 5 1/2 | 24 |
| 7 | 38 5/8 | 44 1/8 | 31 1/4 | 43 | 31 3/4 | 32 1/4 | 30 | 25 1/2 | 40 13/16 | 38 1/4 | 42 | 40 1/2 | 2 1/2 | 6 1/2 | 28 |
| 8 | 43 7/8 | 50 1/8 | 35 1/4 | 48 | 34 3/4 | 36 3/4 | 33 | 27 1/2 | 43 13/16 | 40 3/4 | 48 | 45 | 3 | 7 1/2 | 33 |
| 9 | 49 3/8 | 56 1/8 | 39 5/8 | 54 | 39 3/4 | 40 3/4 | 36 | 30 3/4 | 48 7/16 | 45 3/4 | 54 | 34 3/4 | 19 1/4 | 7 3/4 | 38 |
| 10 | 54 5/8 | 62 1/8 | 43 1/2 | 59 | 42 3/4 | 46 3/4 | 39 | 33 | 51 7/16 | 48 1/2 | 60 | 34 3/4 | 24 1/4 | 8 3/4 | 48 |
| 11 | 59 7/8 | 68 1/8 | 47 1/4 | 64 | 45 3/4 | 51 3/4 | 42 | 34 1/2 | 55 9/16 | 51 | 66 | 40 3/4 | 23 1/4 | 8 3/4 | 54 |
| 12 | 65 1/8 | 74 1/8 | 51 3/8 | 69 | 48 3/4 | 57 3/4 | 45 | 35 3/4 | 58 9/16 | 52 1/4 | 72 | 40 3/4 | 28 1/4 | 8 3/4 | 58 |
| 14 | 76 | 86 1/8 | 59 1/2 | 80 | 54 3/4 | 68 3/4 | 48 | 40 1/4 | 62 9/16 | 59 3/4 | 84 | 41 1/2 | 38 1/2 | 11 3/4 | 86 |
| 16 | 86 5/8 | 98 1/8 | 67 3/8 | 91 | 60 3/4 | 79 3/4 | 51 | 43 3/4 | 65 9/16 | 62 3/4 | 96 | 41 1/2 | 49 1/2 | 11 3/4 | 90 |

| Fan Number | P | Q | R | S | T | U | V | W | X | Y | Z | & | Shaft at Bearing | Shaft at Wheel |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 6 | 14 1/4 | 22 | 18 | 1 3/4 | 7 1/2 | 5 3/4 | 11 1/2 | 13 | 20 1/2 | 19 1/4 | 15 1/4 | 1 | 2 15/16 | 2 15/16 |
| 7 | 17 1/4 | 25 1/2 | 21 1/2 | 1 3/4 | 9 | 5 3/4 | 13 | 13 1/2 | 22 1/4 | 21 3/4 | 17 3/4 | 1 | 3 7/16 | 3 7/16 |
| 8 | 18 7/8 | 28 | 23 | 1 3/4 | 10 1/4 | 5 3/4 | 14 3/4 | 15 1/4 | 25 | 24 | 20 | 1 | 3 7/16 | 3 11/16 |
| 9 | 21 3/8 | 32 | 27 | 3 | 10 1/8 | 7 | 14 1/8 | 19 | 28 3/4 | 23 7/8 | 19 7/8 | 1 | 3 15/16 | 3 15/16 |
| 10 | 22 7/8 | 35 | 30 | 3 | 10 1/8 | 7 | 14 1/8 | 20 1/2 | 30 1/4 | 23 7/8 | 19 7/8 | 1 | 3 15/16 | 4 7/16 |
| 11 | 24 3/8 | 38 1/2 | 34 | 3 | 10 1/8 | 7 | 14 1/8 | 20 1/2 | 30 1/4 | 23 7/8 | 19 7/8 | 1 | 4 7/16 | 4 7/16 |
| 12 | 25 7/8 | 41 1/2 | 36 | 3 | 10 1/8 | 7 | 14 1/8 | 21 1/2 | 31 1/4 | 23 7/8 | 19 7/8 | 1 | 4 7/16 | 4 15/16 |
| 14 | 29 5/8 | 46 3/4 | 40 | 3 | 10 1/8 | 7 | 14 1/8 | 24 1/2 | 34 1/4 | 23 7/8 | 19 7/8 | 1 1/4 | 4 15/16 | 4 15/16 |
| 16 | 32 5/8 | 55 3/4 | 49 | 3 | 10 1/8 | 7 | 14 1/8 | 27 1/2 | 37 1/4 | 23 7/8 | 19 1/3 | 1 1/2 | 4 15/16 | 5 7/16 |

TABLE 108

CAPACITY TABLE FOR THE "ABC" AIR WASHING AND COOLING FANS

| Diameter of Fan | C.F.M. | Outlet Velocity | Static Pressure in Inches, W. C. | | | | | |
|---|---|---|---|---|---|---|---|---|
| | | | ¼-in. | | ½-in. | | ¾-in. | |
| | | | B.hp. | R.p.m. | B.hp. | R.p.m. | B.hp. | R.p.m. |
| 30 | 4,000 | 1000 | .78 | 339 | 1.3 | 420 | 1.89 | 489 |
| 30 | 4,800 | 1200 | 1.11 | 368 | 1.68 | 446 | 2.33 | 512 |
| 30 | 5,600 | 1400 | 1.54 | 402 | 2.18 | 475 | 2.86 | 537 |
| 36 | 4,340 | 1000 | .847 | 282 | 1.41 | 350 | 2.04 | 407 |
| 36 | 5,210 | 1200 | 1.21 | 307 | 1.83 | 372 | 2.54 | 427 |
| 36 | 6,080 | 1400 | 1.67 | 335 | 2.37 | 396 | 3.11 | 448 |
| 42 | 6,680 | 1000 | 1.31 | 341 | 2.18 | 300 | 3.14 | 349 |
| 42 | 8,020 | 1200 | 1.86 | 232 | 2.81 | 318 | 3.91 | 366 |
| 42 | 9,350 | 1400 | 2.56 | 2.7 | 3.64 | 339 | 4.78 | 386 |
| 48 | 8,040 | 1000 | 1.57 | 211 | 2.62 | 262 | 3.78 | 306 |
| 48 | 9,650 | 1200 | 2.23 | 230 | 3.39 | 279 | 4.68 | 320 |
| 48 | 11,250 | 1400 | 3.08 | 251 | 4.37 | 297 | 5.75 | 336 |
| 54 | 10,040 | 1000 | 1.96 | 188 | 3.28 | 233 | 4.74 | 272 |
| 54 | 12,050 | 1200 | 2.79 | 204 | 4.22 | 248 | 5.85 | 284 |
| 54 | 14,050 | 1400 | 3.85 | 223 | 5.47 | 264 | 7.18 | 298 |
| 60 | 11,680 | 1000 | 2.28 | 169 | 3.8 | 210 | 5.5 | 244 |
| 60 | 14,000 | 1200 | 3.24 | 183 | 4.91 | 223 | 6.81 | 256 |
| 60 | 16,350 | 1400 | 4.48 | 200 | 6.37 | 238 | 8.35 | 268 |
| 66 | 14,700 | 1000 | 2.74 | 154 | 4.57 | 191 | 6.63 | 222 |
| 66 | 16,880 | 1200 | 3.89 | 167 | 5.93 | 202 | 8.21 | 233 |
| 66 | 19,700 | 1400 | 5.40 | 182 | 7.67 | 216 | 10.10 | 244 |
| 72 | 16,000 | 1000 | 3.12 | 141 | 5.21 | 175 | 7.54 | 204 |
| 72 | 19,200 | 1200 | 4.44 | 153 | 6.74 | 186 | 9.35 | 213 |
| 72 | 22,400 | 1400 | 6.14 | 167 | 8.71 | 198 | 11.49 | 224 |
| 84 | 20,200 | 1000 | 3.94 | 121 | 6.58 | 150 | 9.52 | 175 |
| 84 | 24,240 | 1200 | 5.62 | 131 | 8.48 | 159 | 11.78 | 183 |
| 84 | 28,300 | 1400 | 7.75 | 143 | 11.00 | 170 | 14.45 | 192 |
| 96 | 25,000 | 1000 | 4.88 | 101 | 8.15 | 131 | 11.77 | 153 |
| 96 | 30,000 | 1200 | 6.94 | 115 | 10.50 | 139 | 14.60 | 160 |
| 96 | 35,000 | 1400 | 9.60 | 125 | 13.60 | 148 | 17.90 | 168 |
| 108 | 33,050 | 1000 | 6.44 | 94 | 10.75 | 117 | 15.55 | 136 |
| 108 | 39,600 | 1200 | 9.16 | 102 | 13.86 | 124 | 19.07 | 142 |
| 108 | 46,300 | 1400 | 12.70 | 112 | 18.00 | 132 | 23.2 | 149 |
| 120 | 39,100 | 1000 | 7.62 | 85 | 12.7 | 105 | 18.4 | 122 |
| 120 | 46,900 | 1200 | 10.85 | 92 | 16.4 | 112 | 22.8 | 128 |
| 120 | 54,700 | 1400 | 15.00 | 100 | 21.3 | 119 | 28.0 | 134 |
| 132 | 45,600 | 1000 | 8.88 | 77 | 14.8 | 96 | 21.45 | 111 |
| 132 | 54,700 | 1200 | 12.65 | 84 | 19.2 | 101 | 26.60 | 116 |
| 132 | 63,800 | 1400 | 17.4 | 91 | 24.80 | 108 | 32.60 | 122 |
| 144 | 52,600 | 1000 | 10.25 | 70 | 17.10 | 88 | 24.75 | 102 |
| 144 | 63,100 | 1200 | 14.60 | 77 | 22.10 | 93 | 30.70 | 107 |
| 144 | 73,600 | 1400 | 20.20 | 84 | 28.62 | 99 | 37.60 | 112 |
| 156 | 60,000 | 1000 | 11.70 | 66 | 19.50 | 81 | 28.21 | 94 |
| 156 | 72,000 | 1200 | 16.67 | 70 | 25.25 | 86 | 35.0 | 99 |
| 156 | 84,000 | 1400 | 23.00 | 77 | 32.70 | 92 | 43.0 | 103 |

These capacities are for Single Inlet Fans. At the same speeds and pressures Double Inlet Fans will deliver twice the volume of air and will require twice the amount of power for operating.

## TABLE 109

### Carrier Type "A" Air Washer

Dimensions and Capacities

| Free Area, Square Feet | Washing Surface, Square Feet | Size of Door, Inches | Gallons per Minute | | | Water Pipes | | Pump | | | | Steam Pipe | | Height | Width | Length | Capacity in Cubic Feet of Air per Minute | Number |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Spray | Flooding | Both | To pump | Fresh | Size | R.p.m. | Horse power | | 0 lb. | 5 lb. | | | | | |
| | | | | | | | | | | Brake | Size of motor | | | | | | | |
| | | | | | | | | | | | | | | Ft. In. | Ft. In. | Ft. In. | | |
| 2.95 | 28.6 | | 9 | 5 | 14 | $1\frac{1}{2}$ | $\frac{3}{4}$ | $1\frac{1}{2}$ | 1700 | .9 | 2 | 1 | 1 | | 1 $5\frac{1}{4}$ | | 1,500 | 1A |
| 6.23 | 60.5 | | 18 | 10 | 28 | $1\frac{1}{2}$ | $\frac{3}{4}$ | $1\frac{1}{2}$ | 1700 | 1.2 | 2 | $1\frac{1}{2}$ | $1\frac{1}{2}$ | | 2 9 | | 3,100 | 2A |
| 9.50 | 92.5 | | 27 | 15 | 42 | 2 | $\frac{3}{4}$ | $1\frac{1}{2}$ | 1700 | 1.5 | 2 | 2 | $1\frac{1}{2}$ | | 4 $0\frac{3}{4}$ | | 4,800 | 3A |
| 12.8 | 124.0 | | 36 | 20 | 56 | 2 | $\frac{3}{4}$ | $1\frac{1}{2}$ | 1700 | 1.8 | 3 | 2 | 2 | | 5 $4\frac{1}{2}$ | | 6,400 | 4A |
| 16.0 | 155.0 | | 45 | 25 | 70 | 2 | $\frac{3}{4}$ | $1\frac{1}{2}$ | 1700 | 2.1 | 3 | $2\frac{1}{2}$ | 2 | | 6 8 | | 8,000 | 5A |
| 19.3 | 187.0 | | 54 | 31 | 85 | 3 | $\frac{3}{4}$ | 2 | 1700 | 2.4 | 3 | $2\frac{1}{2}$ | $2\frac{1}{2}$ | 4 $1\frac{1}{2}$ | 7 $11\frac{3}{4}$ | | 9,700 | 6A |
| 22.6 | 219.0 | | 63 | 36 | 99 | 3 | $\frac{3}{4}$ | 2 | 1700 | 2.7 | 3 | 3 | $2\frac{1}{2}$ | | 9 $3\frac{1}{2}$ | | 11,300 | 7A |
| 25.9 | 251.0 | | 72 | 41 | 113 | 3 | $\frac{3}{4}$ | 2 | 1700 | 3.0 | 5 | 3 | $2\frac{1}{2}$ | | 10 $7\frac{1}{4}$ | | 13,000 | 8A |
| 29.1 | 282.0 | | 81 | 46 | 127 | 3 | 1 | $2\frac{1}{2}$ | 1700 | 3.3 | 5 | 3 | 3 | | 11 11 | | 14,600 | 9A |
| 32.4 | 315.0 | | 90 | 52 | 142 | 3 | 1 | $2\frac{1}{2}$ | 1700 | 3.6 | 5 | $3\frac{1}{2}$ | 3 | | 13 $2\frac{1}{4}$ | | 16,200 | 10A |
| 35.7 | 346.0 | | 99 | 57 | 156 | 3 | 1 | $2\frac{1}{2}$ | 1700 | 3.9 | 5 | $3\frac{1}{2}$ | 3 | | 14 $6\frac{1}{4}$ | | 17,800 | 11A |
| 4.13 | 39.2 | $15\frac{1}{2}\times26$ | 11 | 5 | 16 | 2 | $\frac{3}{4}$ | $1\frac{1}{2}$ | 1700 | 1.0 | 2 | $1\frac{1}{4}$ | 1 | | 1 $5\frac{1}{4}$ | | 2,100 | 1B |
| 8.71 | 84.5 | | 22 | 10 | 32 | 2 | $\frac{3}{4}$ | $1\frac{1}{2}$ | 1700 | 1.3 | 2 | 2 | $1\frac{1}{2}$ | | 2 9 | | 4,400 | 2B |
| 13.3 | 129.0 | | 33 | 15 | 48 | 2 | $\frac{3}{4}$ | $1\frac{1}{2}$ | 1700 | 1.6 | 2 | 2 | $1\frac{1}{2}$ | | 4 $0\frac{3}{4}$ | 7 $2\frac{5}{16}$ | 6,700 | 3B |
| 17.9 | 174.0 | | 43 | 20 | 63 | 2 | $\frac{3}{4}$ | $1\frac{1}{2}$ | 1700 | 1.9 | 3 | $2\frac{1}{2}$ | $2\frac{1}{2}$ | | 5 $4\frac{1}{2}$ | | 9,000 | 4B |
| 22.5 | 218.0 | | 54 | 25 | 79 | $2\frac{1}{2}$ | $\frac{3}{4}$ | 2 | 1700 | 2.3 | 3 | 3 | $2\frac{1}{2}$ | | 6 8 | | 11,300 | 5B |
| 27.1 | 263.0 | | 65 | 31 | 96 | 3 | $\frac{3}{4}$ | 2 | 1700 | 2.6 | 3 | 3 | 3 | | 7 $11\frac{3}{4}$ | | 13,600 | 6B |
| 31.7 | 308.0 | | 76 | 36 | 112 | 3 | $\frac{3}{4}$ | 2 | 1700 | 3.0 | 5 | 3 | 3 | 5 $1\frac{1}{2}$ | 9 $3\frac{3}{4}$ | | 15,800 | 7B |
| 36.2 | 351.0 | | 87 | 41 | 128 | 3 | $\frac{3}{4}$ | $2\frac{1}{2}$ | 1700 | 3.3 | 5 | $3\frac{1}{2}$ | 3 | | 10 $7\frac{1}{4}$ | | 18,100 | 8B |
| 40.8 | 396.0 | | 97 | 46 | 143 | 3 | 1 | $2\frac{1}{2}$ | 1700 | 3.6 | 5 | $3\frac{1}{2}$ | $3\frac{1}{2}$ | | 11 11 | | 20,400 | 9B |
| 45.4 | 440.0 | | 108 | 52 | 160 | 3 | 1 | $2\frac{1}{2}$ | 1700 | 4.0 | 5 | 4 | $3\frac{1}{2}$ | | 13 $2\frac{1}{2}$ | | 22,700 | 10B |
| 50.0 | 485.0 | | 119 | 57 | 176 | 3 | 1 | $2\frac{1}{2}$ | 1700 | 4.4 | 5 | 4 | $3\frac{1}{2}$ | | 14 $6\frac{1}{4}$ | | 25,000 | 11B |
| 54.6 | 530.0 | | 130 | 62 | 192 | 3 | 1 | $2\frac{1}{2}$ | 1700 | 4.7 | $7\frac{1}{2}$ | 4 | 4 | | 15 10 | | 27,300 | 12B |
| 6.40 | 62.0 | 16 ×36 | 18 | 5 | 23 | $1\frac{1}{2}$ | $\frac{3}{4}$ | $1\frac{1}{2}$ | 1700 | 1.1 | 2 | $1\frac{1}{2}$ | $1\frac{1}{4}$ | | 1 $5\frac{1}{4}$ | | 3,200 | 1C |
| 13.7 | 133.0 | | 36 | 10 | 46 | 2 | $\frac{3}{4}$ | $1\frac{1}{2}$ | 1700 | 1.6 | 2 | 2 | 2 | 7 $1\frac{1}{2}$ | 2 9 | | 6,900 | 2C |
| 20.9 | 203.0 | | 54 | 15 | 69 | 2 | $\frac{3}{4}$ | $1\frac{1}{2}$ | 1700 | 2.1 | 3 | $2\frac{1}{2}$ | $2\frac{1}{2}$ | | 4 $0\frac{3}{4}$ | | 10,500 | 3C |

| | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 28.1 | 273.0 | | 72 | 20 | 92 | 3 | ¾ | 2 | 1700 | 2.6 | 3 | 3 | 3 | | 5 | 4½ | | 14,100 | 4C |
| 35.3 | 343.0 | | 90 | 25 | 115 | 3 | ¾ | 2 | 1700 | 3.1 | 5 | 3½ | 3 | | 6 | 8 | | 17,700 | 5C |
| 42.5 | 413.0 | | 108 | 31 | 139 | 3 | ¾ | 2½ | 1700 | 3.6 | 5 | 3½ | 3½ | | 7 | 11¾ | | 21,300 | 6C |
| 49.7 | 483.0 | | 126 | 36 | 162 | 3 | ¾ | 2½ | 1700 | 4.1 | 5 | 4 | 3½ | | 9 | 3½ | | 24,900 | 7C |
| 56.9 | 553.0 | | 144 | 41 | 185 | 3 | ¾ | 2½ | 1700 | 4.6 | 5 | 4 | 4 | | 10 | 7¼ | | 28,500 | 8C |
| 64.1 | 623.0 | | 162 | 46 | 208 | 3 | 1 | 2½ | 1700 | 5.1 | 7½ | 4½ | 4 | | 11 | 11 | | 32,100 | 9C |
| 71.3 | 693.0 | | 180 | 52 | 232 | 4 | 1 | 3 | 1700 | 5.6 | 7½ | 5 | 4½ | | 13 | 2½ | | 35,700 | 10C |
| 78.5 | 763.0 | | 198 | 57 | 255 | 4 | 1 | 3 | 1700 | 6.1 | 7½ | 5 | 4½ | | 14 | 6¼ | | 39,300 | 11C |
| 85.7 | 852.0 | | 216 | 62 | 278 | 4 | 1 | 3 | 1700 | 6.5 | 10 | 5 | 4½ | | 15 | 10 | | 42,800 | 12C |
| 92.9 | 902.0 | | 234 | 67 | 301 | 5 | 1 | 4 | 1120 | 7.1 | 10 | 6 | 5 | 7 1½ | 17 | 1¾ | | 46,500 | 13C |
| 100.0 | 970.0 | | 252 | 73 | 325 | 5 | 1 | 4 | 1120 | 7.5 | 10 | 6 | 5 | | 18 | 5½ | | 50,000 | 14C |
| 107.0 | 1040.0 | | 270 | 78 | 348 | 5 | 1 | 4 | 1120 | 7.9 | 10 | 6 | 5 | | 19 | 9¼ | | 53,500 | 15C |
| 114.0 | 1110.0 | | 288 | 83 | 371 | 5 | 1¼ | 4 | 1120 | 8.3 | 10 | 6 | 6 | | 21 | 1 | | 57,000 | 16C |
| 122.0 | 1180.0 | | 306 | 88 | 394 | 5 | 1¼ | 4 | 1120 | 8.7 | 10 | 6 | 6 | | 22 | 4½ | | 61,000 | 17C |
| 129.0 | 1250.0 | | 324 | 93 | 417 | 5 | 1¼ | 4 | 1120 | 9.0 | 10 | 7 | 6 | | 23 | 8¼ | | 65,000 | 18C |
| 136.0 | 1320.0 | | 342 | 99 | 441 | 5 | 1¼ | 4 | 1120 | 9.4 | 10 | 7 | 6 | | 25 | 0 | | 68,000 | 19C |
| 143.0 | 1390.0 | | 360 | 104 | 464 | 5 | 1¼ | 4 | 1120 | 9.8 | 15 | 7 | 6 | | 26 | 3¾ | | 72,000 | 20C |
| 150.0 | 1460.0 | | 378 | 109 | 487 | 5 | 1¼ | 4 | 1120 | 10.1 | 15 | 7 | 6 | | 27 | 7½ | | 75,000 | 21C |
| 158.0 | 1530.0 | | 396 | 114 | 510 | 5 | 1¼ | 4 | 1120 | 10.5 | 15 | 7 | 7 | | 28 | 11 | | 79,000 | 22C |
| 165.0 | 1600.0 | 16 ×36 | 414 | 120 | 534 | 6 | 1¼ | 5 | 1120 | 10.9 | 15 | 7 | 7 | | 30 | 2¾ | 7 2 5/16 | 83,000 | 23C |
| 8.9 | 86.0 | | 24 | 5 | 29 | 1½ | ¾ | 1½ | 1700 | 1.2 | 2 | 2 | 1½ | | 1 | 5¼ | | 4,400 | 1D |
| 18.7 | 181.0 | | 47 | 10 | 57 | 2 | ¾ | 1½ | 1700 | 1.8 | 3 | 2½ | 2½ | | 2 | 9 | | 9,400 | 2D |
| 28.5 | 276.0 | | 70 | 15 | 85 | 2½ | ¾ | 2 | 1700 | 2.5 | 3 | 3 | 3 | | 4 | 0¾ | | 14,300 | 3D |
| 38.4 | 373.0 | | 94 | 20 | 114 | 3 | ¾ | 2 | 1700 | 3.0 | 3 | 3½ | 3 | | 5 | 4½ | | 19,200 | 4D |
| 48.2 | 468.0 | | 117 | 25 | 142 | 3 | ¾ | 2½ | 1700 | 3.6 | 7½ | 3½ | 3½ | | 6 | 8 | | 24,100 | 5D |
| 58.0 | 563.0 | | 140 | 31 | 171 | 3 | ¾ | 2½ | 1700 | 4.2 | 7½ | 4½ | 4 | | 7 | 11¾ | | 29,000 | 6D |
| 67.8 | 658.0 | | 164 | 36 | 200 | 3 | ¾ | 2½ | 1700 | 4.9 | 7½ | 4½ | 4 | | 9 | 3½ | | 33,900 | 7D |
| 77.6 | 754.0 | | 187 | 41 | 228 | 4 | ¾ | 3 | 1700 | 5.5 | 7½ | 5 | 4½ | | 10 | 7¼ | | 38,800 | 8D |
| 87.4 | 847.0 | | 210 | 46 | 256 | 4 | 1 | 3 | 1700 | 6.1 | 10 | 5 | 5 | | 11 | 11 | | 43,700 | 9D |
| 97.2 | 940.0 | | 234 | 52 | 286 | 4 | 1 | 3 | 1700 | 6.7 | 10 | 6 | 5 | 9 1½ | 13 | 2½ | | 48,600 | 10D |
| 107.0 | 1040.0 | | 258 | 57 | 315 | 5 | 1 | 4 | 1120 | 7.2 | 10 | 6 | 5 | | 14 | 6¼ | | 53,500 | 11D |
| 117.0 | 1140.0 | | 281 | 62 | 343 | 5 | 1 | 4 | 1120 | 7.8 | 10 | 6 | 6 | | 15 | 10 | | 59,000 | 12D |
| 127.0 | 1230.0 | | 304 | 67 | 371 | 5 | 1 | 4 | 1120 | 8.3 | 10 | 6 | 6 | | 17 | 1¾ | | 64,000 | 13D |
| 137.0 | 1330.0 | | 328 | 73 | 401 | 5 | 1 | 4 | 1120 | 8.7 | 15 | 7 | 6 | | 18 | 5½ | | 69,000 | 14D |
| 146.0 | 1420.0 | | 352 | 78 | 430 | 6 | 1 | 5 | 1120 | 9.2 | 15 | 7 | 6 | | 19 | 9¼ | | 73,000 | 15D |
| 156.0 | 1520.0 | | 375 | 83 | 458 | 6 | 1¼ | 5 | 1120 | 9.7 | 15 | 7 | 7 | | 21 | 1 | | 78,000 | 16D |
| 166.0 | 1610.0 | | 398 | 88 | 486 | 6 | 1¼ | 5 | 1120 | 10.1 | 15 | 7 | 7 | | 22 | 4½ | | 83,000 | 17D |
| 176.0 | 1710.0 | | 422 | 93 | 515 | 6 | 1¼ | 5 | 1120 | 10.6 | 15 | 8 | 7 | | 23 | 8¼ | | 88,000 | 18D |
| 186.0 | 1800.0 | | 445 | 99 | 544 | 6 | 1¼ | 5 | 1120 | 11.0 | 15 | 8 | 7 | | 25 | 0 | | 93,000 | 19D |

TABLE 109—*Continued*

| Free Area, Square Feet | Washing Surface, Square Feet | Size of Door, Inches | Gallons per Minute | | | Water Pipes | | Pump | | | | Steam Pipe | | Height | Width | Length | Capacity in Cubic Feet of Air per Minute | Number |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Spray | Flooding | Both | To pump | Fresh | Size | R.p.m. | Horse power | | 0 lb. | 5 lb. | | | | | |
| | | | | | | | | | | Brake | Size of motor | | | | | | | |
| | | | | | | | | | | | | | | Ft. In. | Ft. In. | Ft. In. | | |
| 195.0 | 1900.0 | | 468 | 104 | 572 | 6 | $1\frac{1}{4}$ | 5 | 1120 | 11.5 | 15 | 8 | 7 | | 26 $3\frac{3}{4}$ | | 98,000 | 20D |
| 205.0 | 1990.0 | | 492 | 109 | 601 | 6 | $1\frac{1}{4}$ | 5 | 1120 | 11.9 | 15 | 8 | 8 | | 27 $7\frac{1}{2}$ | | 103,000 | 21D |
| 215.0 | 2090.0 | | 515 | 114 | 629 | 6 | $1\frac{1}{4}$ | 5 | 1120 | 12.4 | 15 | 8 | 8 | 9 $1\frac{1}{2}$ | 28 11 | | 108,000 | 22D |
| 225.0 | 2190.0 | | 538 | 120 | 658 | 6 | $1\frac{1}{4}$ | 5 | 1120 | 12.9 | 15 | 8 | 8 | | 30 $2\frac{3}{4}$ | | 113,000 | 23D |
| 11.2 | 109.0 | | 29 | 5 | 34 | $1\frac{1}{2}$ | $\frac{3}{4}$ | $1\frac{1}{2}$ | 1700 | 1.3 | 2 | 2 | 2 | | 1 $5\frac{3}{4}$ | | 5,600 | 1E |
| 23.6 | 229.0 | | 58 | 10 | 68 | $2\frac{1}{2}$ | $\frac{3}{4}$ | $1\frac{1}{2}$ | 1700 | 2.0 | 3 | 3 | $2\frac{1}{2}$ | | 2 $9\frac{1}{2}$ | | 11,800 | 2E |
| 36.1 | 350.0 | | 87 | 15 | 102 | 3 | $\frac{3}{4}$ | 2 | 1700 | 2.8 | 5 | $3\frac{1}{2}$ | 3 | | 4 $1\frac{1}{4}$ | | 18,100 | 3E |
| 48.6 | 472.0 | | 115 | 20 | 135 | 3 | $\frac{3}{4}$ | $2\frac{1}{2}$ | 1700 | 3.5 | 5 | 4 | $3\frac{1}{2}$ | | 5 5 | | 24,300 | 4E |
| 61.0 | 592.0 | | 144 | 25 | 169 | 3 | $\frac{3}{4}$ | $2\frac{1}{2}$ | 1700 | 4.2 | 5 | $4\frac{1}{2}$ | 4 | | 6 $8\frac{1}{2}$ | | 31,000 | 5E |
| 73.4 | 712.0 | | 173 | 31 | 204 | 3 | $\frac{3}{4}$ | $2\frac{1}{2}$ | 1700 | 5.0 | $7\frac{1}{2}$ | 5 | $4\frac{1}{2}$ | | 8 $0\frac{1}{4}$ | | 36,700 | 6E |
| 85.8 | 833.0 | | 202 | 36 | 238 | 4 | $\frac{3}{4}$ | 3 | 1700 | 5.7 | $7\frac{1}{2}$ | 5 | $4\frac{1}{2}$ | | 9 4 | | 42,900 | 7E |
| 98.2 | 953.0 | | 230 | 41 | 271 | 4 | $\frac{3}{4}$ | 3 | 1700 | 6.4 | $7\frac{1}{2}$ | 6 | 5 | | 10 $7\frac{3}{4}$ | | 49,100 | 8E |
| 110.0 | 1070.0 | | 259 | 46 | 305 | 5 | 1 | 4 | 1120 | 7.1 | 10 | 6 | 6 | | 11 $11\frac{1}{2}$ | | 55,000 | 9E |
| 123.0 | 1190.0 | 16 × 36 | 288 | 52 | 345 | 5 | 1 | 4 | 1120 | 7.8 | 10 | 6 | 6 | | 13 3 | 7 $2\frac{5}{16}$ | 62,000 | 10E |
| 135.0 | 1310.0 | | 317 | 57 | 374 | 5 | 1 | 4 | 1120 | 8.3 | 10 | 7 | 6 | | 14 $6\frac{3}{4}$ | | 68,000 | 11E |
| 148.0 | 1430.0 | | 346 | 62 | 408 | 5 | 1 | 4 | 1120 | 8.9 | 10 | 7 | 6 | 11 $1\frac{3}{4}$ | 15 $10\frac{1}{2}$ | | 74,000 | 12E |
| 160.0 | 1550.0 | | 375 | 67 | 442 | 5 | 1 | 4 | 1120 | 9.4 | 10 | 7 | 7 | | 17 $2\frac{1}{4}$ | | 80,000 | 13E |
| 173.0 | 1680.0 | | 404 | 73 | 477 | 5 | 1 | 4 | 1120 | 10.0 | 15 | 7 | 7 | | 18 6 | | 87,000 | 14E |
| 185.0 | 1800.0 | | 432 | 78 | 510 | 5 | 1 | 4 | 1120 | 10.5 | 15 | 8 | 7 | | 19 $9\frac{3}{4}$ | | 93,000 | 15E |
| 198.0 | 1920.0 | | 462 | 83 | 545 | 6 | $1\frac{1}{4}$ | 5 | 1120 | 11.0 | 15 | 8 | 7 | | 21 $1\frac{1}{2}$ | | 99,000 | 16E |
| 210.0 | 2040.0 | | 490 | 88 | 578 | 6 | $1\frac{1}{4}$ | 5 | 1120 | 11.5 | 15 | 8 | 8 | | 22 5 | | 105,000 | 17E |
| 222.0 | 2160.0 | | 518 | 93 | 611 | 6 | $1\frac{1}{4}$ | 5 | 1120 | 12.1 | 15 | 8 | 8 | | 23 $8\frac{3}{4}$ | | 111,000 | 18E |
| 235.0 | 2280.0 | | 547 | 99 | 646 | 6 | $1\frac{1}{4}$ | 5 | 1120 | 12.6 | 15 | 10 | 8 | | 25 $0\frac{1}{2}$ | | 118,000 | 19E |
| 241.0 | 2400.0 | | 576 | 104 | 680 | 6 | $1\frac{1}{4}$ | 5 | 1120 | 13.2 | 15 | 10 | 8 | | 26 $4\frac{1}{2}$ | | 124,000 | 20E |
| 260.0 | 2520.0 | | 605 | 109 | 714 | 6 | $1\frac{1}{4}$ | 5 | 1120 | 13.7 | 15 | 10 | 8 | | 27 8 | | 130,000 | 21E |
| 273.0 | 2650.0 | | 634 | 114 | 748 | 6 | $1\frac{1}{4}$ | 5 | 1120 | 14.2 | 15 | 10 | 10 | | 28 $11\frac{1}{2}$ | | 137,000 | 22E |
| 285.0 | 2770.0 | | 663 | 120 | 783 | 6 | $1\frac{1}{4}$ | 5 | 1120 | 14.7 | 20 | 10 | 10 | | 30 $3\frac{1}{2}$ | | 143,000 | 23E |

## CHAPTER XVI

# MISCELLANEOUS APPLICATIONS OF REFRIGERATION

Some of the applications of refrigeration are very special, and it seems best to bring these all together into one chapter. In some of the following there is no established method of solution, while in others, for example, in the case of the problem in shaft-sinking, refrigeration has been abandoned, practically, as a modern method, but some of the devices outlined, however, may be useful in solving other problems not mentioned specifically in the text.

### SKATING RINKS

The skating rink is one of the popular uses to which mechanical refrigeration has been applied. Although it would seem at first that it is similar in most respects to other applications, yet in reality it is very different, in fact unique, in the details of design and operation. Of course, there is the factor of the making of ice, and yet it is entirely different from commercial ice making, inasmuch as skating rinks require an ice surface of particular quality and temperature and nothing else. It has been found that the best surface for skating is one that is very nearly at 32 deg. F., but which is not wet on the surface and has at the same time a temperature of from 22 to 24 deg. F. near the refrigerating coils. A room temperature of from 48 to 50 degrees is comfortable and is the most desirable for skaters.

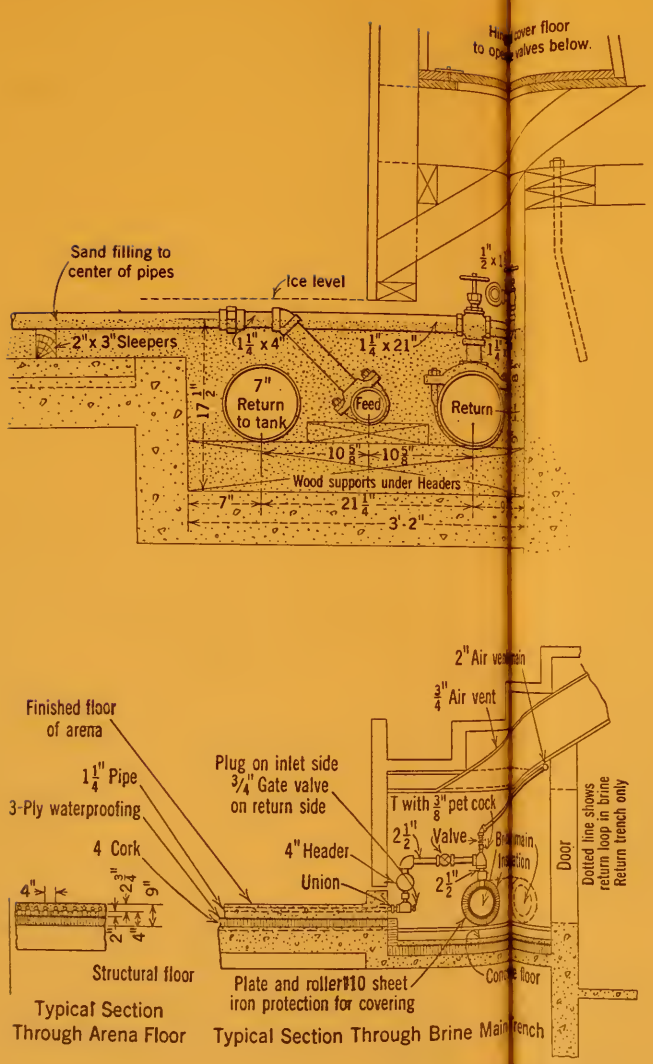
**Design of Skating Rinks.**—The skating floor should be as solid as possible. Unless concrete is used it is well to have about 10 to 12 in. of sand on which are laid stringers on about 24 to 36-in. centers to which the piping is attached. The piping can be either 1 in. or $1\frac{1}{4}$ in., erected on from 3 to 6-in. centers, but 4 to $4\frac{1}{2}$-in. centers will give the best results, as there is less likelihood of soft places between the pipes if the larger spacing is not used. The brine system is the one used almost entirely because of the freedom from variable temperatures, and because of the ability with brine to store up refrigeration in a brine tank for the occasion of a peak load which is almost inevitable in the case of the

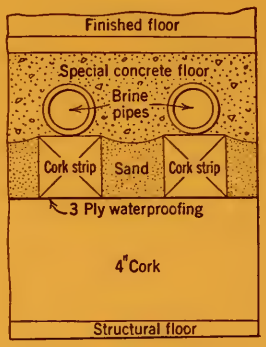
skating rink. A brine temperature of about 16 to 18 deg. F. gives the most satisfactory results, as it has been found that if the ice is too cold, and hard, it does not wear well but forms snow easily—a condition that does not exist with the ice at or near 32 degrees.

The brine lines should be arranged so as to be readily accessible. There is nothing standard in regard to the supports for the pipes. The Vancouver, B. C., rink is built on lumber, while the Duquesne Garden was built on a wooden floor with some three or four feet of open space beneath it. The Winter Garden, Pittsburgh, is laid entirely, and the Seattle rink, partly, on concrete. The Chicago Arena, the Cleveland, the Portland (Oregon), and the Victoria rinks were laid directly on the floor. The essential is that the floor be level. Insulation of the floor is considered unnecessary and simply an added expense.

There is no standard method of piping. Some rinks are designed for a reversal of brine flow and others are not. As a rule the brine headers run the long way of the rink and the pipes are arranged to run across the short dimension. Some engineers specify that the piping shall be laid out so that each separate circuit is of equal length in order that the regulation of flow can be obtained without throttling the shorter circuits. The amount of brine flow varies considerably also, but a brine temperature rise of from two to four degrees only, should be permitted, and the temperature during the operation of the rink should be approximately from 16 to 18 degrees. It is essential that brine piping be laid out so that air pockets cannot be formed and so that the air be carried upwards to a point where air cocks can be placed. The resistance to the flow of the brine should not be greater than 4 to 6 lb. per sq. in. in order to keep the pumping cost of operation as low as practical. A ratio of 1 gal. of brine per minute to 23 sq. ft. of floor area was used with success in the Chicago Arena.

As regards methods of calculation of the tonnage required, it will be more satisfactory to refer to the table showing what has been done before in the case of rinks that have been successful. Some tests by D. H. Scott at the Cleveland rink showed that a heating effect of 21,000,000 B.t.u. per day prevailed at the time of the test. Of this amount it was estimated that 18 per cent was due to direct sunlight, 12 per cent to heat leakage through the walls and the roof, 3 per cent to air changes, 20 per cent to requirements in freezing the ice, 9 per cent to a special temperature control used at this rink, 8 per cent to the heat evolved by the skaters, 1 per cent to the electric illumination, leaving 29 per cent unaccounted for. It would seem that the loss due to infiltration would be more than 3 per cent, and some heat would be given by the ground. As the skating season extends from the fall through the



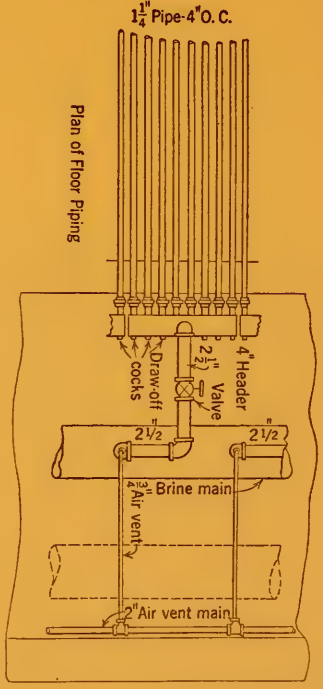




FIG. 339.—Skating Rink Details.

*To face page 547.*

winter into the spring, it is out of the question to make any careful analysis of the refrigeration requirements, and it seems much better to be governed by designs in successful operation. It has been usual to

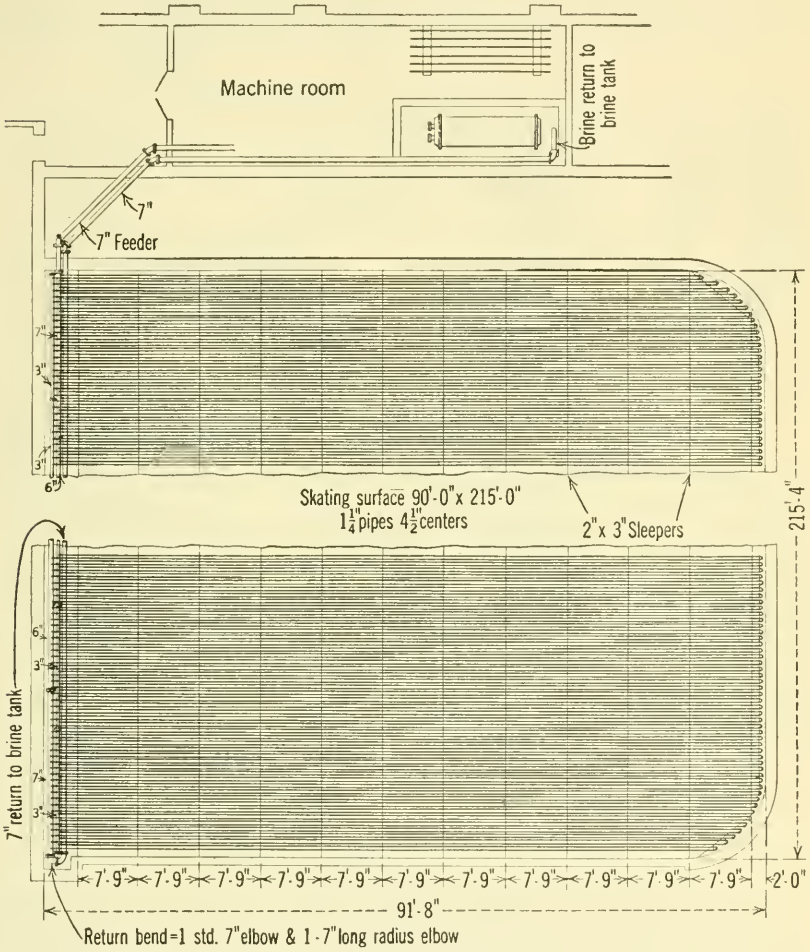



Fig. 338.—Skating Rinks.

design the piping so that the floor may be used for other purposes in the summer time.

The average cost of rinks has been $1.40 per sq. ft. of ice surface. The usual allowance is 20 sq. ft. of ice surface per skater, which is considered liberal, and 12 sq. ft. is considered to give a crowded condition. Figures 338 and 339 give typical arrangements of brine piping.

TABLE 110

| | Ice Surface, Square Feet | Feet of 1-In. Pipe | Feet of $1\frac{1}{4}$-In. Pipe | Linear Feet per Square Foot of Floor | System |
|---|---|---|---|---|---|
| Madison Square Garden | 15,820 (85×186) | ..... | 53,000 | 3.35 | Brine |
| Pittsburgh | 15,750 | 72,000 | ..... | 4.57 | Direct exp. |
| St. Nicholas | 14,400 | ..... | 44,000 | 3.06 | Brine |
| Chicago (arena) | 33,900 | 106,200 | ..... | 3.13 | Brine |
| Berlin | 26,900 | ..... | 82,000 | 3.05 | Brine |
| Washington, D. C. | 23,600 | ..... | 96,000 | 4.06 | Direct exp. |
| Duluth, Mich. | 19,350 (90×215) | ..... | 52,000 | 2.69 | Brine |

| | Ice Surface, Square Feet | Tonnage | Square Feet of Ice Surface per Ton |
|---|---|---|---|
| Duquesne Garden | 23,600 (90×262) | 135 | 175 |
| Brooklyn | 15,285 | 80 | 191 |
| St. Nicholas | 14,400 | 80 | 180 |
| Washington, D. C. | 23,600 | 100 | 236 |
| Spokane | 16,000 | 65 | 246 |
| Toronto | 16,000 | 100 | 160 |
| Cleveland | 20,500 (86×240) | 130 | 158 |
| Chicago (arena) | 33,925 (115×295) | 215 | 158 |
| Portland, Oregon | 27,300 (85×321) | 175 | 156 |

## THE CHILLING OF CASTINGS

In the chilling of castings, such as in the manufacture of plowshares and mold boards, one of the objects of the treatment is to secure a uniform glass hardness. The chilling may be accomplished by the use of calcium chloride brine in the case of iron castings, but a quenching oil is used in the case of steel manufacture. It is understood that if water were used in either case the heat of the metal would generate considerable steam, and while the greater part of this would relieve itself readily, yet some would cling to the surface of the metal and form a surface film of gas. The action of such a surface film is to cause resistance to the transfer of heat, and the metal would be cooled non-uniformly and soft

and hard spots would be formed. The action of brine is different as the gas film does not seem to be so marked as in the case of the water. If oil is used a slower cooling results because of the lower value of the coefficient of heat transfer from the metal to the oil than from the metal to the brine. Experience, however, has shown that sodium chloride only

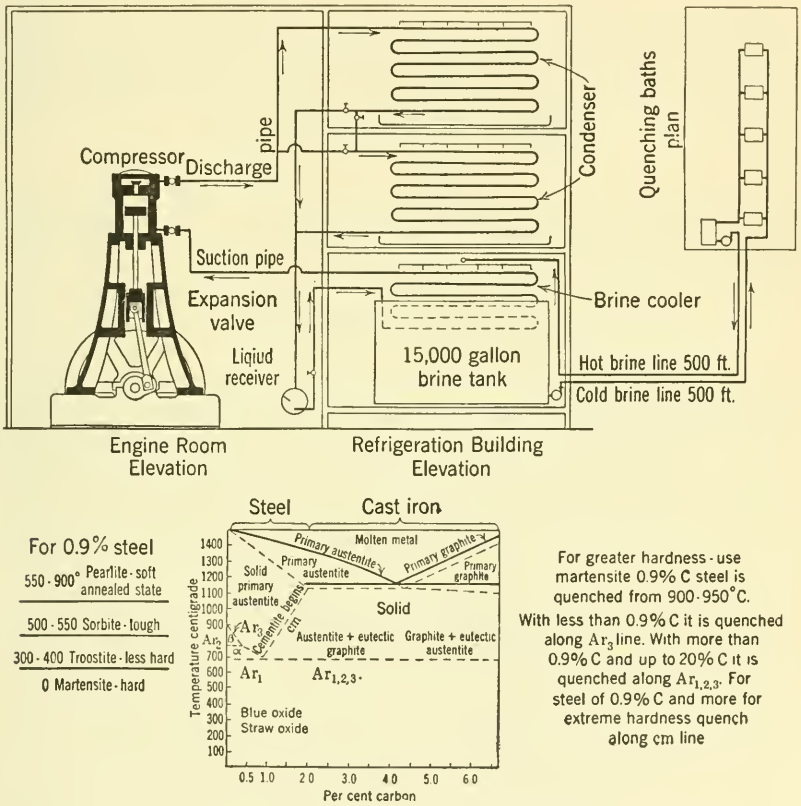



Fig. 340.—The Chilling of Cast Iron.

is the salt giving the required glass hardness expected in the manufacture of plowshares. The following is taken from the John Deere Plow Company records:

**Problem.**—The daily run consists of 45,500 lb. of castings which enters the brine bath at 1650 deg. F. and is cooled to 50 degrees (in 9 hours). The brine is to to be cooled to 45 deg. F. and is to be heated 10 degrees only. Find the amount of refrigeration required and the amount of brine to be circulated per minute.

The specific heat of the casting is to be taken as 0.169, and of the brine of a density of 1.115 as 0.843. The refrigeration required is

$$\frac{45500}{9} \times (1650 - 50) \times 0.169 = 1{,}367{,}000 \text{ B.t.u. per hour.}$$

$$= 113.9 \text{ tons}$$

$$\text{Adding 20 per cent for unavoidable losses} = 22.8$$

$$= 136.7 \text{ tons}$$

Referring to Fig. 340 it will be seen that the brine lines are quite long so that a liberal factor needs to be used for the losses. The brine required per minute will be:

$$\frac{1{,}640{,}000}{60} = M \times 0.843 \times (55 - 45).$$

$$M = 3242 \text{ lb. per minute.}$$

Taking the density of the brine at 1.115, the number of gallons to be circulated per minute will be 349.

## THE POETSCH PROCESS

A construction process used some years ago, but which has been neglected of late years, is the freezing process of sinking vertical shafts in aqueous soils and quicksand. The essential feature of the process is to sink pipes for brine or direct expansion sufficiently close together to enable the water in the coil to be frozen solid. For this purpose brine at about — 15 deg. F. is taken and the pipes are located about 3 ft. apart.

An estimate of the refrigeration required can be made when the details are known. Frequently the specific gravity of the dry solids is taken at 1.8, the specific heat at 0.2, and the average water content in the earth at 25 lb. per cu. ft. The frozen ring may extend $1\frac{1}{2}$ ft. outside, and 3 ft. inside the center of the line of pipes, but the ground is cooled 5 ft. beyond the frozen zone. For calculating the time of freezing, an average value of 85 B.t.u. per sq. ft. of pipe surface is recommended, and 33 per cent is added to the total calculated refrigeration for losses. The total time of freezing usually is from 5 to 6 weeks.

## THE PRODUCTION OF VERY LOW TEMPERATURES

The manufacture of liquid air cannot be said to be an application of refrigeration, although at times some use of refrigerating processes is made during a part of the course of operations. For example, in one plant of the Linde Air Products Company, which has a capacity of 55,000 cu. ft. of free air per hour, a $3\frac{1}{2}$-in. by 10-in. $CO_2$ compressor is

used in the fourth stage after-cooler in order to separate out, by cooling the air to − 20 deg. F., as much moisture as possible at this point rather than to permit it to continue on to the expansion valve where frost would accumulate and interfere with the continuity of action. Liquid air may be produced by two principal methods: the Linde and the Claude processes.

In the Linde process the atmospheric air is purified by passing it through a lime scrubber in order to reduce the carbon dioxide content in the air to 0.003 per cent, which would also freeze on the throttle valve were it permitted to continue in the liquefaction process, and the air is then compressed. In some of the plants 55 per cent of the total capacity of the plant is compressed in four stages to 2000 lb. per sq. in., and 45 per cent is compressed to 70 lb. gage, as this ratio appears to work satisfactorily. Cylinder lubrication is secured by means of a soap solution. The depression of the temperature at the throttle valve is obtained by means of the Joule-Thompson effect,[1] and by a very carefully designed counterflow exchanger and rectification column (Fig. 341). The depression of the temperature being cumulative, the formation of liquid air finally commences at the throttle valve and floods the lower coils (the vaporizer), and by boiling performs the function of refrigerating the compressed gas in these coils to the point of liquefaction. The liquid air passes upward through $d_2$ to the top of the rectification column where it trickles down through the screenings, etc., of the column. The vaporized liquid from the vaporizer passes counter-current to the descending liquid air, and in due time an equilibrium is established with a differential temperature of about 20 deg. F. (− 194 deg. C. at the top and − 183 deg. C. at the bottom). As nitrogen boils at − 196 deg. C. and oxygen at − 183 deg. C., the result of this process is that the nitrogen gas, with 7 per cent oxygen, is formed at the top and oxygen, with 0.8 per cent nitrogen gas at the bottom. The remainder of the apparatus is designed to provide the counterflow required to cool the entering compressed gas and to heat the extremely cold gases leaving the coils.

The result of the Linde process is the production of nearly pure oxygen, but the nitrogen has about 7 per cent of oxygen present. Bayley's

[1] The Joule-Thompson effect for air is about 0.5 deg. F. per atmosphere drop of pressure during throttling. The depression of temperature (in deg. C.) is given by the expression

$$t_1 - t_2 = 0.276(p_1 - p_2)\left(\frac{273}{T}\right)^2$$

where $p_1$ and $p_2$ are the pressures before and after throttling, in atmospheres, and $T$ is the initial temperature in deg. C. abs.

experiments in 1900 proved that this percentage of oxygen could not be decreased when using the rectification column.

The Claude process uses a combination of an expansion engine for two-thirds of the gas and a throttling valve for the remaining one-third. The gas leaving the expansion engine is at about four atmospheres and is at or near the liquefaction temperature point, whereas the air which arrives at the expansion valve is liquid at about 33 atmospheres. A selective system of rectification is employed which is rendered possible by a preliminary partial separation of nitrogen and oxygen and a combined system of heat interchange and drying which eliminates the necessity of a chemical removal of moisture from the air. Figure 341

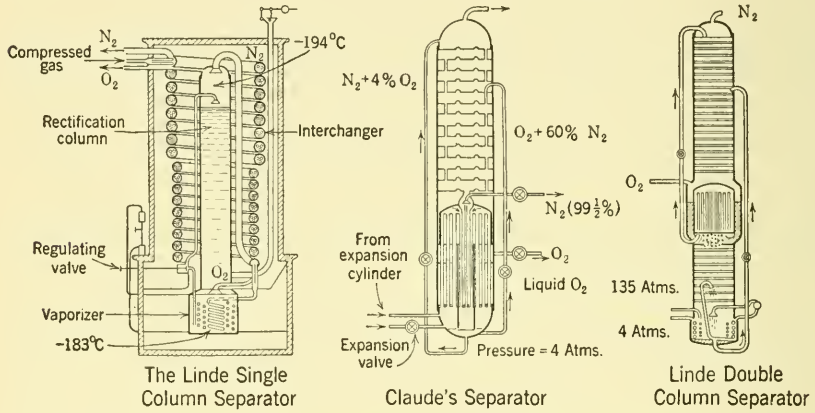



Fig. 341.—The Production of Very Low Temperatures.

shows the Claude system details to some extent. The power required in liquid air plants is approximately 30 to 33 b.hp. per 1000 cu. ft. of oxygen per hour.

## THE ICE CREAM AND MILK INDUSTRY

Before the general use of mechanical refrigeration, the manufacture of ice cream and the pasteurizing of milk and cream were limited to the use of ice and salt for cooling. The result of the introduction of mechanical refrigeration has been a very rapid extension of the industry during the last twenty years. The ice cream manufactured previous to 1900 in the United States is estimated as at about 25 million gallons yearly, with few concerns of capacities greater than 100,000 gallons, whereas the capacity in 1920 was 260 million gallons yearly, and a number of

firms had outputs greater than a million gallons. The number of refrigerating machines and their capacities had doubled from 1919 to 1922. The advance in the use of mechanical refrigeration has not been so rapid in the dairy plants. The listed plants were of 9300 tons of refrigeration in 1904, 16,800 tons in 1914, and 46,400 tons in 1922, with an average tonnage per machine of 13.5 tons.

**Pasteurizing of Milk.**—Quick cooling of milk is an absolute necessity when the milk is not consumed immediately. Referring to Table 111

TABLE 111

Effect of Time and Temperature on the Growth of Bacteria in Milk

| Temperature | 24 Hours | 48 Hours | 96 Hours | 168 Hours |
|---|---|---|---|---|
| 32 deg. F. ( 0 deg. C.) | 2,400 | 2,100 | 1,850 | 1,400 |
| | **30,000** | **27,000** | **24,000** | **19,000** |
| 39 deg. F. ( 4 deg. C.) | 2,500 | 3,600 | 218,000 | 4,200,000 |
| | **38,000** | **56,000** | **4,300,000** | **38,000,000** |
| 42 deg. F. ( 5 deg. C.) | 2,600 | 3,600 | 400,000 | |
| | **43,000** | **210,000** | **5,760,000** | |
| 46 deg. F. ( 6 deg. C.) | 3,100 | 12,000 | 1,480,000 | |
| | **42,000** | **360,000** | **12,200,000** | |
| 50 deg. F. (10 deg. C.) | 11,600 | 540,000 | | |
| | **89,000** | **1,940,000** | | |
| 55 deg. F. (13 deg. C.) | 18,800 | 3,400,000 | | |
| | **187,000** | **38,000,000** | | |
| 60 deg. F. (16 deg. C.) | 180,000 | 28,000,000 | | |
| | **900,000** | **168,000,000** | | |
| 68 deg. F. (20 deg. C.) | 450,000 | 25,000,000,000 | | |
| | **4,000,000** | **25,000,000,000** | | |
| 86 deg. F. (30 deg. C.) | 1,400,000,000 | | | |
| | **14,000,000,000** | | | |
| 94 deg. F. (35 deg. C.) | 25,000,000,000 | | | |
| | **25,000,000,000** | | | |

good quality milk held at 42 deg. F. for 24 hours showed 2400 bacilli, whereas if held for the same time at 86 degrees the same cubic centimeter of milk would contain 1,400,000,000 bacilli. When milk is consumed in cities and large towns it frequently is between 36 and 48 hours old when delivered to the consumer, and it is likely to be from 48 to 60 hours old when it is consumed. On the other hand it has been found that milk cannot be frozen[2] with satisfactory results. It is evident, then,

[2] In Bulletin No. 98, United States Department of Agriculture (By J. T. Bowen), it says:

"According to Kasdorf, when raw milk was partly frozen at a temperature of

that raw milk is dangerous for all purposes other than for cooking except under the most sanitary conditions of production and bottling unless some very drastic method of curtailment or sterilization of the bacilli is possible. The present method is to "pasteurize" the milk (and cream) by bringing the temperature up to a point which will kill them or stunt their growth. This has been found to be achieved with satisfaction by, first, heating to about 165 deg. F. for a half-minute (the flash minute), and, second, of heating to only 140 degrees[3] and holding the temperature for 30 minutes (Fig. 342). The latter is the more usual method, as there is less heat to be removed from the milk and less likelihood of a scorched taste to the milk. After passing through the pasteurizing process the milk or cream must be cooled quickly to 40 deg. F. This is done by the so-called Baudalot cooler, an atmospheric type of cooler, or the regenerator type of cooler (Fig. 345), which permits a counterflow of the cold raw milk and the hot pasteurized milk. In the Baudalot cooler (Fig. 344), the first part of the cooling is done

10.5 deg. F., in the ordinary container, during transportation it was found that ice first formed around the sides and at the bottom of the cans; the central core contained most of the casein, sugar, and other mineral ingredients, while most of the fat was found in the top layer of the liquid portion.

"When milk which has been frozen gradually, without agitation, is thawed out clots will be found floating in the liquid (composed mostly of albumen and fat, which may be dissolved by cooking); on the other hand, if the milk is preserved in a frozen condition for three or four weeks, these clots will be very hard to dissolve, and the difficulty experienced in dissolving them increases in proportion to the length of time the milk is preserved in a frozen state. For this reason the freezing of milk for the purpose of transportation has so far been very little practiced.

"If the milk is held at 32 deg. F. for a few days some type of bacteria will grow and multiply slowly. With a good quality of milk, i.e., that containing few bacteria, it may take weeks or even months for them to gain great headway. What few bacteria develop at low temperatures are of different species from those ordinarily found at the higher temperatures, and they may produce marked changes in the chemical composition of the milk without specially changing its appearance. Consequently, it is unsafe to assume that milk which has been held for several days at a low temperature is in good condition. According to Pennington, milk exposed continually to a temperature of 29 to 32 deg. F. causes after a lapse of from 7 to 21 days the formation of small ice crystals which gradually increase until the milk is filled with them, and there may be an adherent layer on the walls of the vessel. The milk does not freeze solid. In spite of the fact that the milk was a semi-solid mass of ice crystals, an enormous increase of bacterial content took place. Though the bacterial content was numerically in the *hundreds of millions* per cubic centimeter, there was neither taste nor odor to indicate that such was the case. Neither did the milk curdle when heated, and the unfitness of the milk for household purposes would not ordinarily be detected until the lactic acid bacteria decreased in numbers and the putrefractive bacteria began to develop."

[3] Tests at Macdonald College (Ag. Gazette, Canada, 1917), claim that 152 deg. F. is required to make coli bacteria non-virulent.

with water, and the brine performs the cooling from about 80 degrees to the final temperature of 40 degrees. Too low a final temperature is not desired as the milk is likely to foam during bottling. After bottling, the cases of bottles are stored in a cooler room in order to hold the temperature until such time as delivery takes place.

The problem of refrigeration in the cooling of milk and cream is very simple. The problem is one of cooling a liquid only—from about 80 deg. to 40 deg. F. The only distinctive point is that the milk usually is a short period load, that is, it consumes some 3–5 hours per day and usually is a decided "peak" load. There are in most dairies other

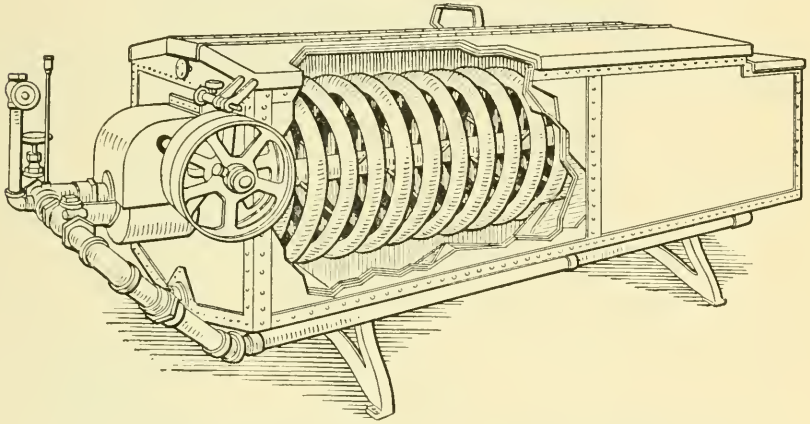


Fig. 342.—The Pasteurizing of Milk.

refrigerating needs in the same plant: for example, milk holding rooms, ice making, etc. The design could be worked out so as to carry the ice making and the storage room by one machine and the milk cooling by another. Frequently, however, the problem works out nicely by the combination with a brine tank, and by cooling the brine down to 10 deg. F. or lower sufficient refrigeration is stored up so that a smaller compressor can be used, and the machine may be operated for 8 to 12 or more hours out of the 24. Such a brine storage is well illustrate l by the following problem:

**Problem.**—Find the refrigeration required to cool 5000 gal. of milk in three (3) hours from 75 to 40 deg. F. by the use of brine. Also find the size of brine storage required to enable the compressor to be operated for ten hours with the pasteurizer in use three (3) hours only. Take an average value of 0.9 (Fig. 343) for the specific heat of the milk and the specific gravity of the milk at 1.032 (Table 112).

The refrigeration required is

$$M \times c \times (t_2 - t_1) = 5000 \times 8\tfrac{1}{3} \times 1.032 \times 35 \times 0.9 = 1{,}355{,}000 \text{ B.t.u.}$$

Adding 10 per cent for losses = 135,500 B.t.u.

= 1,490,500 B.t.u.

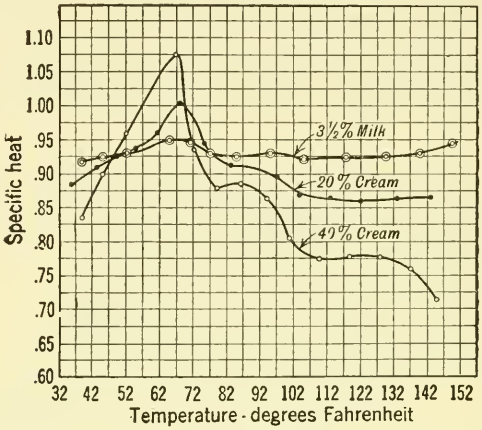



Fig. 343.—The Specific Heat of Milk and Cream.

TABLE 112

Specific Gravity of Milk and Cream Corresponding to Various Percentages of Butter Fat at 68 Deg. F.

| Percentage of Fat | Specific Gravity | Percentage of Fat | Specific Gravity | Percentage of Fat | Specific Gravity | Percentage of Fat | Specific Gravity |
|---|---|---|---|---|---|---|---|
| 0.025 | 1.037 | 11 | 1.024 | 21 | 1.012 | 31 | 1.003 |
| 1 | 1.036 | 12 | 1.022 | 22 | 1.011 | 32 | 1.002 |
| 2 | 1.035 | 13 | 1.020 | 23 | 1.010 | 33 | 1.001 |
| 3 | 1.034 | 14 | 1.019 | 24 | 1.009 | 34 | 1.000 |
| 4 | 1.032 | 15 | 1.018 | 25 | 1.008 | 35 | .999 |
| 5 | 1.031 | 16 | 1.017 | 26 | 1.008 | 36 | .999 |
| 6 | 1.030 | 17 | 1.016 | 27 | 1.007 | 37 | .998 |
| 7 | 1.029 | 18 | 1.015 | 28 | 1.006 | 38 | .997 |
| 8 | 1.027 | 19 | 1.014 | 29 | 1.005 | 39 | .996 |
| 9 | 1.026 | 20 | 1.013 | 30 | 1.004 | 40 | .995 |
| 10 | 1.025 | | | | | | |

Therefore the capacity of the compressor is $1,490,500 \div (10 \times 12,000) = 12.4$ tons. If such a sized compressor was in use the brine storage would have to carry what the machine could not deliver during the 3 hours that the milk is pasteurized for

$$1,490,500 - (12.4 \times 3 \times 12,000) = 1,044,000 \text{ B.t.u.}$$

If brine at zero degrees is carried at the time that the milk cooling began and it was permitted to allow the brine to rise to 20 degrees then the average specific heat would be (for calcium chloride) 0.71 for 1.2 specific gravity, and the volume required would be:

$$Q = M \times c \times (t_2 - t_1)$$

$$1,044,000 = M \times 0.71 \times (20 - 0).$$

$$M_{\text{lb. of brine}} = 1,044,000 \div 14.2 = 73,500 \text{ lb.}$$

$$\text{Volume of brine} = 73,500 \div (62.5 \times 1.2) = 980 \text{ cu. ft.}$$

To store this volume of brine a cylindrical tank 11 ft. 0 in. diameter and 13 ft. 1 in. high would give the volume required, but for safety one 11 ft. diameter by 14 ft. high should be used.

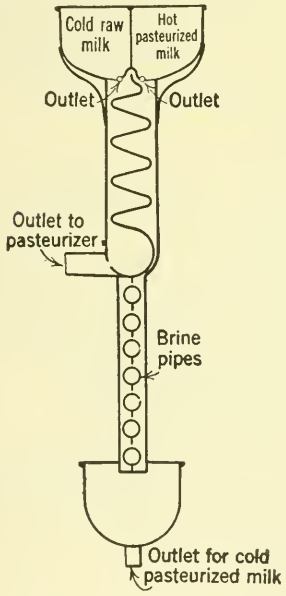



Fig. 344.—The Baudalot Milk Cooler.

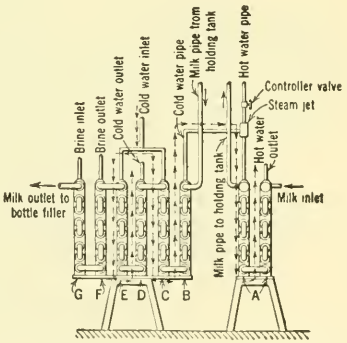
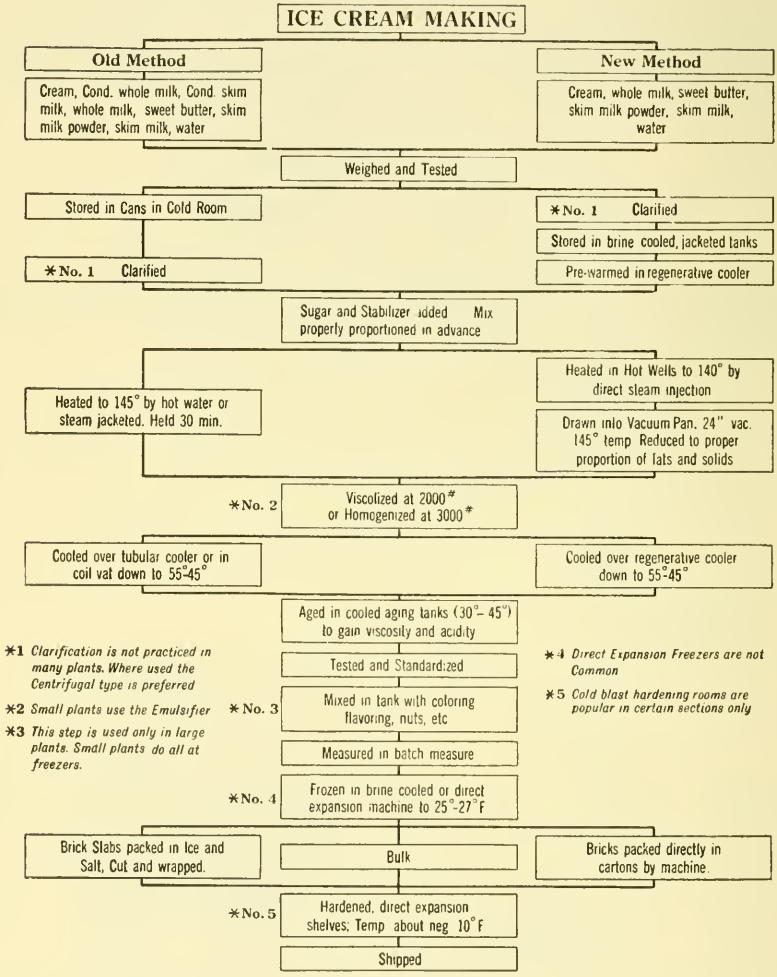



Fig. 345.—The Double Pipe Pasteurizer and Cooler.

The ice required in a milk plant does not have to be transparent, nor does it have to be any particular size or shape. Sometimes the brine tank can be utilized to manufacture ice, and 50 lb. cans will freeze in the 10 hours of operation, but the additional load should be allowed

for, using the ratio of about 200 B.t.u. of refrigeration per 1 lb. of ice manufactured.

TABLE 113.


ICE CREAM MAKING
Old Method
Cream, Cond. whole milk, Cond. skim milk, whole milk, sweet butter, skim milk powder, skim milk, water
New Method
Cream, whole milk, sweet butter, skim milk powder, skim milk, water
Weighed and Tested
Stored in Cans in Cold Room
*No. 1 Clarified
*No. 1 Clarified
Stored in brine cooled, jacketed tanks
Pre-warmed in regenerative cooler
Sugar and Stabilizer added Mix properly proportioned in advance
Heated to 145° by hot water or steam jacketed. Held 30 min.
Heated in Hot Wells to 140° by direct steam injection
Drawn into Vacuum Pan, 24" vac. 145° temp Reduced to proper proportion of fats and solids
*No. 2
Viscolized at 2000# or Homogenized at 3000#
Cooled over tubular cooler or in coil vat down to 55°-45°
Cooled over regenerative cooler down to 55°-45°
Aged in cooled aging tanks (30°- 45°) to gain viscosity and acidity
Tested and Standardized
*No. 3
Mixed in tank with coloring flavoring, nuts, etc
Measured in batch measure
*No. 4
Frozen in brine cooled or direct expansion machine to 25°-27°F
Brick Slabs packed in Ice and Salt, Cut and wrapped.
Bulk
Bricks packed directly in cartons by machine.
*No. 5
Hardened, direct expansion shelves; Temp about neg 10°F
Shipped
*1 Clarification is not practiced in many plants. Where used the Centrifugal type is preferred
*2 Small plants use the Emulsifier
*3 This step is used only in large plants. Small plants do all at freezers.
*4 Direct Expansion Freezers are not Common
*5 Cold blast hardening rooms are popular in certain sections only

The additional refrigeration consists of the storage room loads. When the milk is placed in the cooler it can be chilled somewhat, but not very much unless the room is heavily piped and good circulation of the air is possible. The main desire is to keep the commodity at approximately 40 degrees until the time arrives for loading on the delivery wagons. The milk bottles are placed in cans usually $18\frac{1}{2}$ in. by $14\frac{1}{2}$ in.

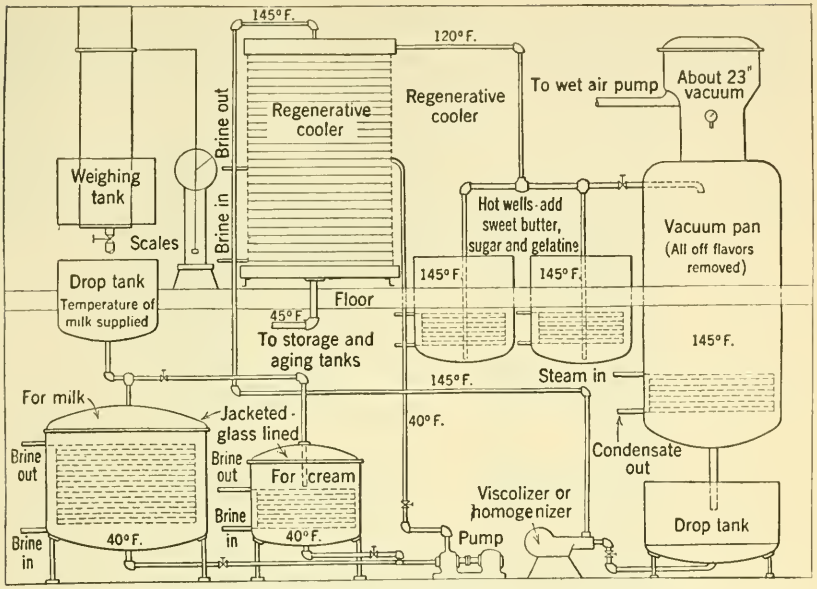



FIG. 346.—The Manufacture of Ice Cream.

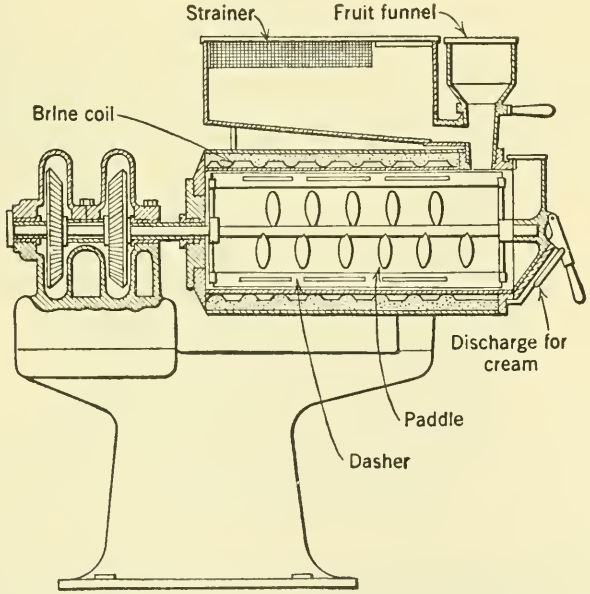



FIG. 347.—The Power Driven Ice Cream Freezer.

by 10 in. high, which hold 12 qts. or 20 pints. The weight of the case is about 14 lb., the weight of 12 one-quart bottles is 22 lb., and the weight of the milk in 12 bottles is 24 lb., making a total of about 60 lb. The

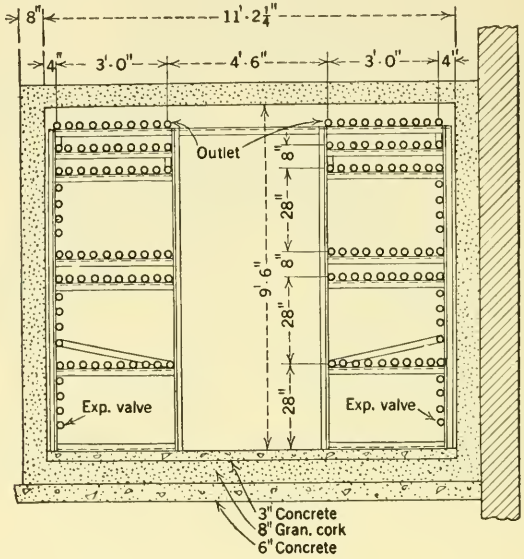



Fig. 348.—Sharp Freezer for Ice Cream.

specific heat of glass is 0.2 and of yellow pine is about 0.6. The glass milk bottle is 4 in. diameter by $9\frac{3}{8}$ in. high for quarts and $3\frac{1}{8}$ in. diameter by $7\frac{1}{4}$ in. high for pints. These last constants may be used in calculating the storage room load.

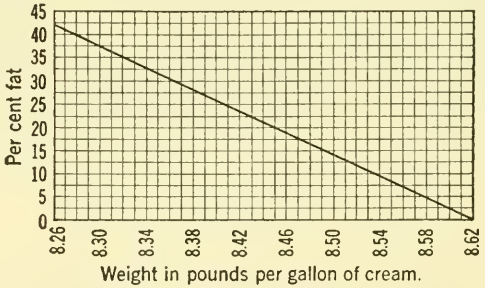



Fig. 349.—Weight of Cream.

**Ice Cream Manufacture.**—Although the process of the manufacture of ice cream is somewhat similar to that of ice making, yet the details are much more complicated and numerous. Table 113 shows the process,

and Fig. 346 gives an idea of the steps in preparing the mix before the freezing process. In brief, the freezing is accomplished by means of power-driven horizontal freezers only (Fig. 347). These may be 40 to 100-qt. capacity, and may be brine cooled with brine from 0 to 5 or 10 deg. F., or by means of direct expansion, using pressures from 15 to 20 lb. gage, but brine is much more easily controlled. The mix enters the freezer at about 40 to 45 degrees and is beaten by the paddles, absorbs

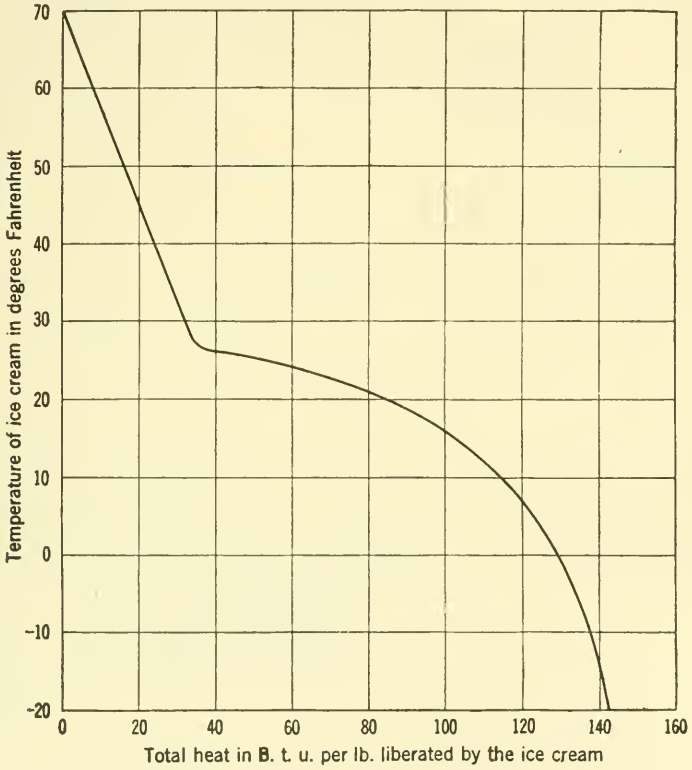



Fig. 350.—Total Heat of Ice Cream.

air, and during the freezing process at 29 to 27 deg. F. swells (acquires an *overrun*) from 50 to 100 per cent (Fig. 352), depending on the operating care during the freezing. The time of freezing is from 10 to 15 minutes, and when removed from the freezer the mix has a consistency of a viscous oil, or a heavy syrup, and is packed in large cylindrical containers of $2\frac{1}{2}$ to 5 gal. capacity, after which they are placed in hardening rooms at 0 to − 10 degrees for the rest of the freezing process. The hardening room (Fig. 348) is usually constructed with the piping

arranged as shelving in the proportion of 1 ft. of $1\frac{1}{4}$-in. pipe per $\frac{3}{4}$ cu. ft. of room space. Still air is usually employed, with an allowance of about 150 to 400 cu. ft. per 100 gal. ice cream capacity, and a corresponding floor area of 15 to 35 sq. ft. The time of storage is usually from $1\frac{1}{2}$ to 3 days.

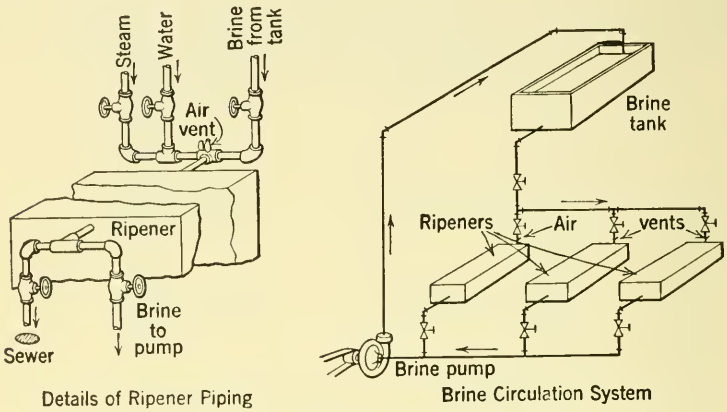



Fig. 351.—Brine Circulation in a Creamery.

The refrigeration required cannot be more than an approximation as ice cream is more like a frozen custard or pudding than anything else, and in consequence the ingredients vary greatly. It has been the custom to assume that the total amount of heat absorbed by the brine during the freezing process took place at constant temperature in the freezer,

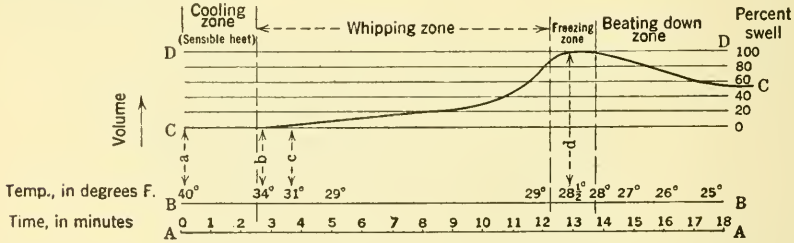



Fig. 352.—The Swell of Ice Cream.

but according to H. F. Zoller[4] careful tests indicate that for a standard vanilla ice cream of 10 per cent milk fat, 14 per cent cane sugar, $10\frac{1}{2}$ per cent milk solids not fat, and 0.5 per cent gelatine the total heat curve is as shown in Fig. 350. Here, freezing of the water content starts at

[4] H. F. Zoller, A. S. R. E., 1924, Freezing and Hardening of Ice Cream.

27 degrees and continues down to − 20 degrees. The heat removed per pound of the mix by the brine or the ammonia in cooling from one

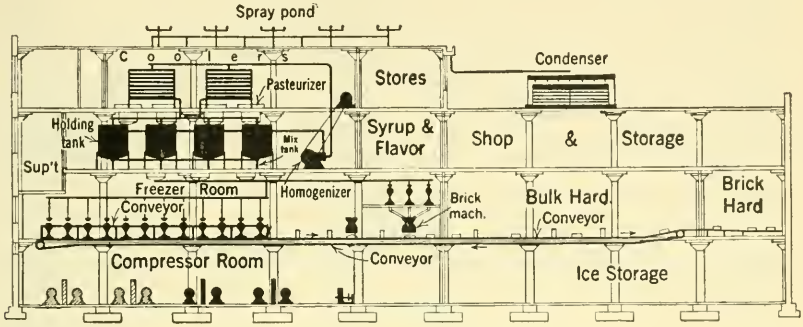



Fig. 353.—Ice Cream Manufacture.

temperature to another is found by subtracting the values of the *ordinates* of these points. For example, if the mix enters the freezer at

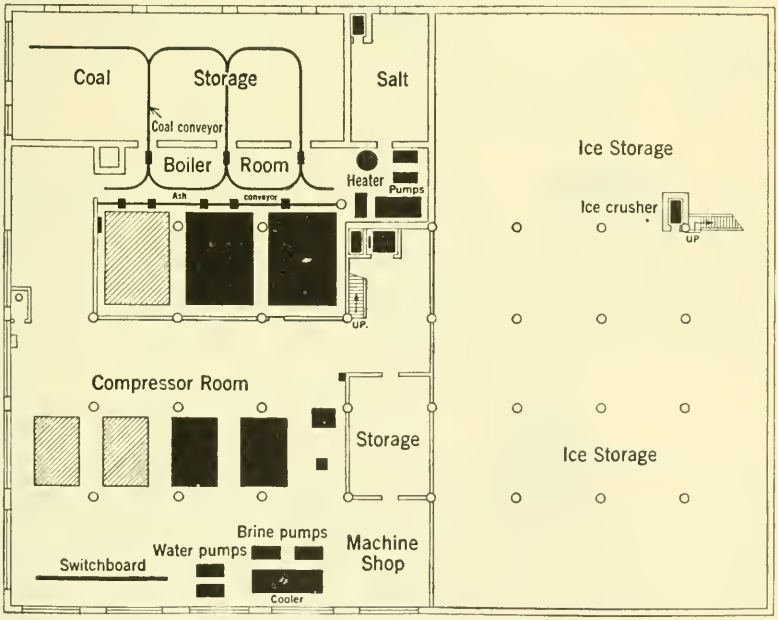



Fig. 354.—Ice Cream Manufacture.

50 degrees and leaves at 21 degrees, the heat absorbed by the refrigerant is 80 − 16 = 64 B.t.u. per lb. If the half-frozen cream is now

placed in the sharp freezer and is lowered to 0 degrees, then the heat removed in the hardening room is 130 − 80 = 50 B.t.u. per lb.

**Problem.**—1000 gal. of ice cream is to be manufactured per day. The swell will be assumed to be 60 per cent. The mix enters the freezers at 40 degrees and the ice cream is kept stored in a hardening room kept at 0 deg. F.

If 1000 gal. is the amount of the finished product, then the amount of the mix is 1000 ÷ 1.6 = 625 gals. If the specific gravity of the mix can be taken as 1.10, then the weight per gallon would be 5.7 + lb. (of the product) and the total weight would be 1000 × 5.72 = 5720 lb. The refrigeration required of the brine in the

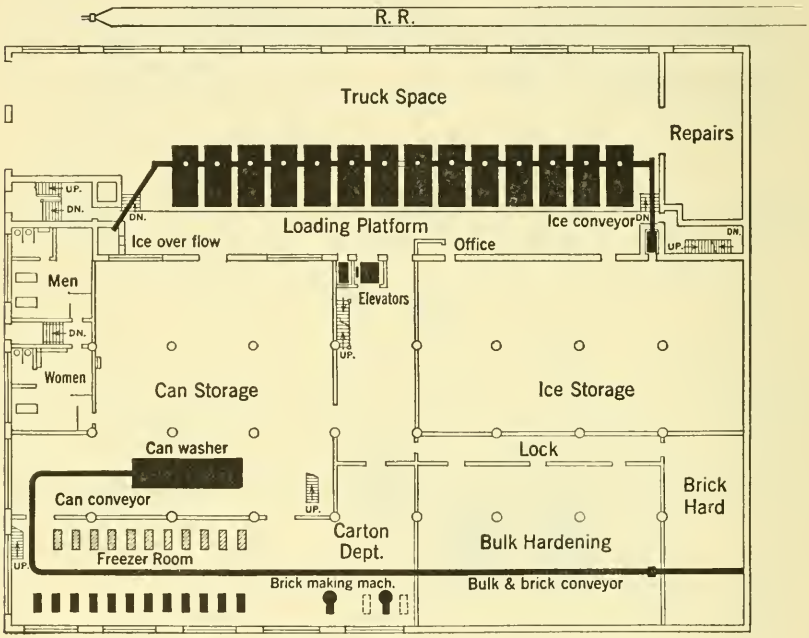



Fig. 355.—Ice Cream Manufacture.

freezers, if the mix enters the freezers at 40 deg. F., would be (80 − 24) × 5720 = 320,300 B.t.u. The brine for the purpose would be at from 0 deg. to 10 deg. F. Adding 10 per cent for leakage and other losses gives a total of 352,300 B.t.u. of refrigeration required of the freezers per 8 hours. The refrigerating work in the hardening room would be (130 − 80) × 5720 = 286,000 B.t.u.,. or 314,600 B.t.u. with the same allowance for losses. If the cream is packed in $2\frac{1}{2}$ and 5-gal. cylindrical containers, these measure 9 in. diameter by 13 in. high and weigh $11\frac{1}{2}$ lb. for the smaller and 9 in. diameter by 23 in. high and weigh 19 lb. for the larger. A hardening room of about the dimensions shown in Fig. 346 would need to be 10 ft. deep for this amount of cream, and the dimensions of the room would be 11 ft. $2\frac{1}{4}$ in. × 10 × 9 ft. 6 in. high inside. With 8-in. corkboard all around and 90 degrees outside tem-

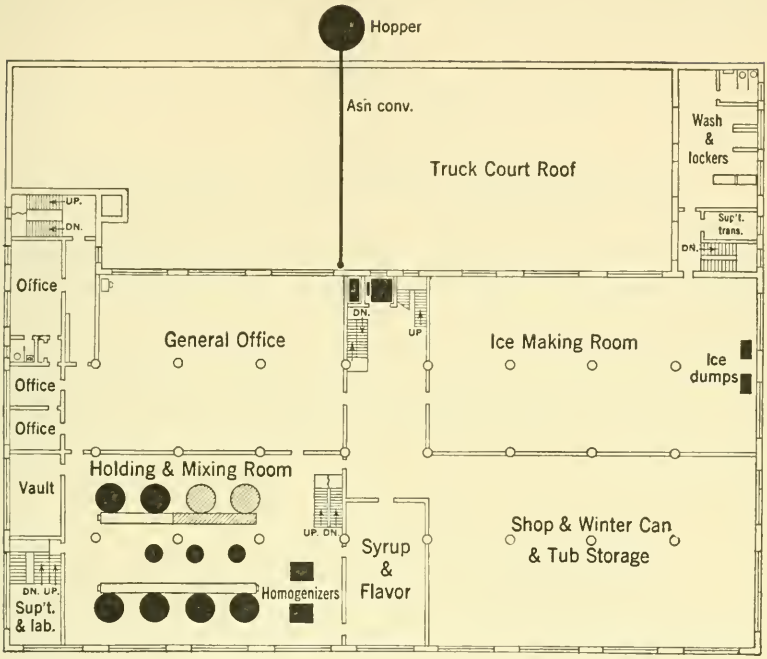



FIG. 356.—Ice Cream Manufacture.

FIG. 357.—Ice Cream Manufacture.

perature of the air and − 10 degrees inside, the heat leakage is, without allowing for the insulating value of the floor:

$$U = \frac{1}{\frac{1}{1.4} + \frac{1}{1.4} + \frac{8\,0}{0.296}} = 0.0352.$$

Therefore,

$$Q = 622.9 \times 0.0352 \times [90 - (-10)] = 2200 \text{ B.t.u. per hour.}$$

Such an excessive temperature difference would not be found during the night time except under very unusual conditions. Frequently the insulation will hold the hardening room during the night time without more than 3 to 5 degrees rise of temperature.

Figure 351 gives some details of brine circulation in the creamery, and Figs. 353 to 357 give the general arrangement in a well-designed ice cream factory. The latter has been designed with provision for expansion as well as an economical routing of the material and finished products.

TABLE 114

VANILLA ICE CREAM, OVER RUN 100 PER CENT

(Initial Temperature of Mix 70 deg. F.)

| | Before Whipping | After Whipping Refrigeration Shut Off | After 2 Days in Hardening Room | After 7 Days in Hardening Room |
|---|---|---|---|---|
| Amount of water frozen into ice (per cent) | 40.0 | 20.0 | 65.0 | 94.0 |
| B.t.u. of refrigeration per lb. of ice cream absorbed by the mix (total) | 92.0 | 78.0 | .... | 138.0 |

## CHAPTER XVII

## HOTEL AND APARTMENT REFRIGERATION

Mechanical refrigeration is now a necessity in the modern hotel to the same extent that the ice cooled box was a few years ago. The larger hotel or club has a large number of special boxes requiring refrigeration (Table 115), located convenient to the workroom or where service or storage is required. These boxes may be quite small, or of fair size, depending on the kind of box. In addition, some hotels make their own ice, calculated from the ratio of 1.6 to 2 tons of refrigeration per ton of ice making, their own ice cream and cool the drinking water supplied to the rooms and dining rooms. The refrigerating load is somewhat indefinite even after the details of the cold storage boxes and other details are known, but an estimate can be made with some degree of precision. For example, the refrigeration for the small boxes can be estimated on the liberal basis of 5.0 B.t.u. per sq. ft. per degree difference of temperature per 24 hours for the heat leakage, with 50 per cent added for the opening of doors, whereas the larger boxes may have a loss due to leakage of only 2.0 B.t.u. and the large cold storage boxes only 1.0 B.t.u. per sq. ft. per degree difference of temperature per 24 hours. These are considered to be very liberal values, in order to take into account the possible poor insulation and the probable poor maintenance of good conditions. As a rule the piping is large by about 25 per cent or more, and less piping will be found to work satisfactorily. For small boxes up to 500 cu. ft. capacity it is usual to use 1-in. pipe, and from 500 to 2000 cu. ft. $1\frac{1}{4}$-in. pipe is usual.

As an example of the use of the table and the chart for the requirements for hotel refrigeration as applied to refrigerator boxes the pantry box will be taken. This is marked B 36, which refers to a box in frequent use to be maintained at a temperature of 36 deg. F. The piping ratio of one such of 500 cu. ft. capacity (Fig. 285) is given as 1.5 cu. ft. per 1 lin. ft. of 1-in. brine pipe, or 334 ft. of pipe will be required. If the outside temperature is taken as 80 deg. F. and the outside area of the box is 390 sq. ft., taking a value of 2.5 for the coefficient of heat transfer of the walls of the box, then the leakage will be

$$Q = 390 \times 2.5 \times (80 - 36) = 42{,}900 \text{ B.t.u. per 24 hours,}$$

to which will be added 50 per cent, or 21,450 B.t.u., per 24 hours for the live load, and the total will be 64,350 B.t.u., or 0.223 tons. The amount of piping taken from the chart, 334 lin. ft. of 1-in. pipe, will give a heat transfer of

$$\frac{334}{2.90} \times 2.0 \times 24 \times (36 - 20) = 88{,}450 \text{ B.t.u. per 24 hours,}$$
$$= 0.307 \text{ ton.}$$

TABLE 115

HOTEL AND RESTAURANT BOXES

| | Rating | Temperature | | Rating | Temperature |
|---|---|---|---|---|---|
| *Baker Shop:* | | | *Kitchen* (Con.): | | |
| Ice cream | A | 28 | Serving box | B | 36 |
| Pantry | B | 36 to 38 | Serving pantry | B | 36 |
| Pastry | B | 36 | Terrapin kitchen | B | 38 |
| | | | Vegetables | C | 38–40 |
| *Café and Restaurant:* | | | Sundries | B | 36 |
| Lunch box | B | 38 | Pickle box | C | 38 |
| Pantry | B | 36 | Salad (serving pantry) | C | 36 |
| Refrigerating box: | | | | | |
| Nurses, officers, | | | *In Storage Room:* | | |
| private | B | 38 | Butter, small | B | 36 |
| | | | Cheese, serving box | B | 36 |
| *Kitchen:* | | | Cigars | C | 36 |
| Corned beef | C | 36 | Cream | B | 36 |
| Pastry counter | B | 36 | Eggs, serving box | B | 31 |
| Dried fish | C | 36 | Fruit | B | 38 |
| Fresh fish | A | 25 to 30 | Milk | B | 36 |
| Freezer | A | 15 to 18 | Pantry | B | 38 |
| Garde manger | B | 36 to 38 | Receiving room | B | 36 |
| Meat-cooler | C | 38 | Storage box | B | 36 |
| Meat cut | B | 35 | Storage box | C | 36 |
| Order box chef | B | 36 | | | |
| Oysters, live | C | 43 to 45 | *Sundries:* | | |
| Oysters, in tubs | B | 36 | Carafe | A | 28 |
| Pantry | B | 36 | Ice storage | A | 26 |
| Poultry and Game | A | 28 | Swill | B | 36 |

Taking the curves for the cubic feet of room space per ton of refrigeration (Fig. 285), the value given for a 500-cu. ft. capacity box, and 36 deg. F., is 1400 cu. ft. per ton of refrigeration, the tonnage becomes

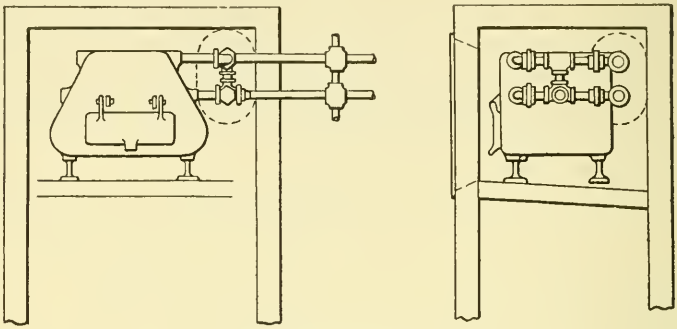
$\frac{500}{1400} = 0.36$. This value, 0.36 ton, is larger than either of the calculated values of tonnage in the preceding paragraph, and it indicates that the tables and curves are calculated liberally and are conservative, so as to allow for all commercial factors of construction and operation. It is usually agreed that direct expansion piping is likely to give less heat transfer than brine piping will, because the ammonia (or carbon dioxide) in evaporating partly fills the inside of the pipe with gas, thereby making the surface inefficient for heat transfer.

The total requirements for hotel refrigeration can be obtained only by a careful computation of the separate cold storage boxes and their operating conditions and all the other uses of refrigeration. Proper allowance must be made for pipe line and other losses.

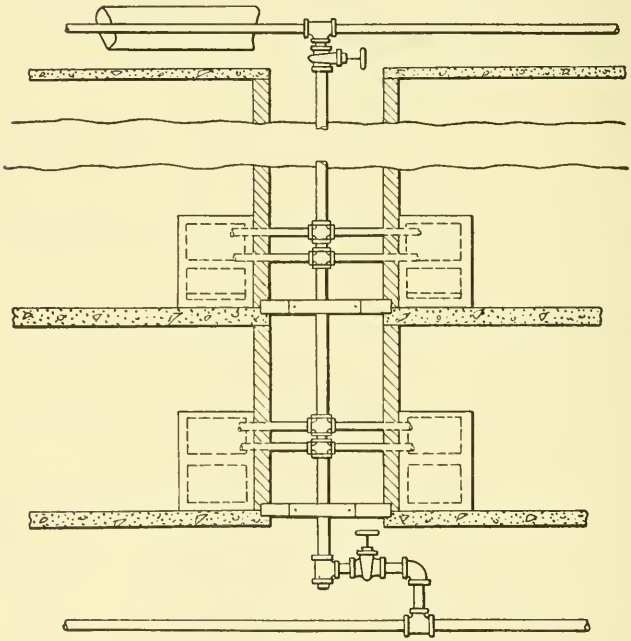
**Apartment Refrigeration.**—All modern apartments of any size are provided with mechanical refrigeration, and this is almost always by means of brine rather than by direct expansion because of better control and because of the greater danger from ammonia were that to be used. It should be noted, however, that carbon dioxide has been popular in this application because of the lack of danger should it happen that a heavy leak should occur. The usual dimensions for apartment boxes vary from 18 in. by 20 in. by 30 in. to 22 in. by 44 in. by 60 in., with 6 cu. ft. and 30 cu. ft. displacement respectively, but the average displacement is 15 cu. ft., corresponding to a box 18 in. by 30 in. by 48 in. The external area of these boxes varies from 20 to 65, with an average of about 40 sq. ft.

**The Refrigeration Required.**—In calculating the total refrigeration required some idea of the details of the installation is necessary. Without question a better grade of refrigerator now is used, i.e., one constructed with 1 in. to 2 in. corkboard. Such tests as are available indicate a leakage loss through the construction of from 0.15 to 0.18, with the 2-in. and the 1-in. corkboard boxes respectively, B.t.u. per sq. ft. per degree difference per hour. The cooling surfaces installed must overcome the heat leakages and the live load due to the opening of the doors and the insertion of the hot and warm commodities. Such surfaces are calculated on the basis of 1.7 B.t.u. as the value of $k$. If pipe coils are used, the $\frac{3}{4}$-in. pipe is the more usual, but cast iron sections or cast iron "brine boxes" are becoming popular, as well as welded boxes made up of sheet iron. Most of these brine boxes (snow mounds) are designed to hold small ice cube pans so as to make small amounts of ice (Fig. 358). The brine temperature carried is usually 15 deg. F. when ice is made, and from 20 to 25 degrees in other cases. The temperature in the box is usually, when brine refrigerated, kept at about 40 degrees.

**Piping.**—The brine piping for apartment refrigeration is usually arranged as shown in Figs. 361, 169 and 362, so as to be balanced. The



Typical Snowmound Connections

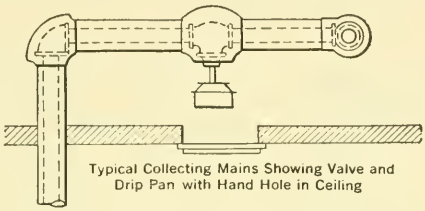


Typical Apartment Riser

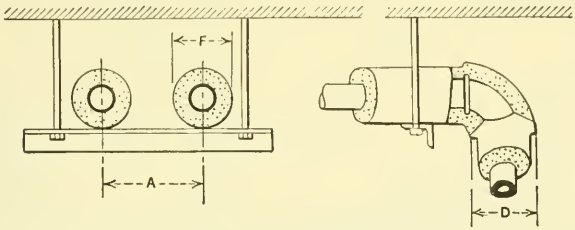
Fig. 358.—Typical Apartment Piping.

shell and coil, the shell and tube and the double pipe brine cooler may be used, with or without brine storage tanks. If no storage tank is used it is necessary to operate the compressor whenever refrigeration is

required as the amount of the brine in the system is small. The double-riser system allows brine of practically the same temperature to enter each box, and the system is not limited to any maximum number of boxes per riser, but the first cost is much greater because of this greater amount of piping. The single riser is popular, especially in apartments,



Typical Collecting Mains Showing Valve and Drip Pan with Hand Hole in Ceiling




Minimum Dimensions

| Size of pipe | A | D | F |
|---|---|---|---|
| ½" | 6¼" | 4½" | 4¼" |
| ¾" | 6¾" | 5" | 4¾" |
| 1" | 7⅜" | 5⅞" | 5⅝" |
| 1¼" | 8⅜" | 6⅝" | 6⅜" |
| 1½" | 9" | 7¼" | 6⅞" |
| 2" | 9½" | 7½" | 7¼" |
| 2½" | 11" | 8⅛" | 7⅞" |
| 3" | 12" | 9⅛" | 8⅞" |
| 3½" | 13" | 10" | 9⅝" |

Typical Hanger and Dimensions to Allow Application of Cork Covering

Fig. 359.—Typical Hanger and Details.

as about half the amount of piping and pipe covering is used, and one-half of the cost of erection, space occupied and the amount of joints with their possible leaks will occur. But with the single-pipe system (Fig. 362) all the brine is required to pass through each box, unless it is by-passed, and the last box receives the highest temperature brine. The single riser is usually limited to 12 to 14 boxes per riser. A brine velocity of not more than 5 ft. per second should be used, and not more

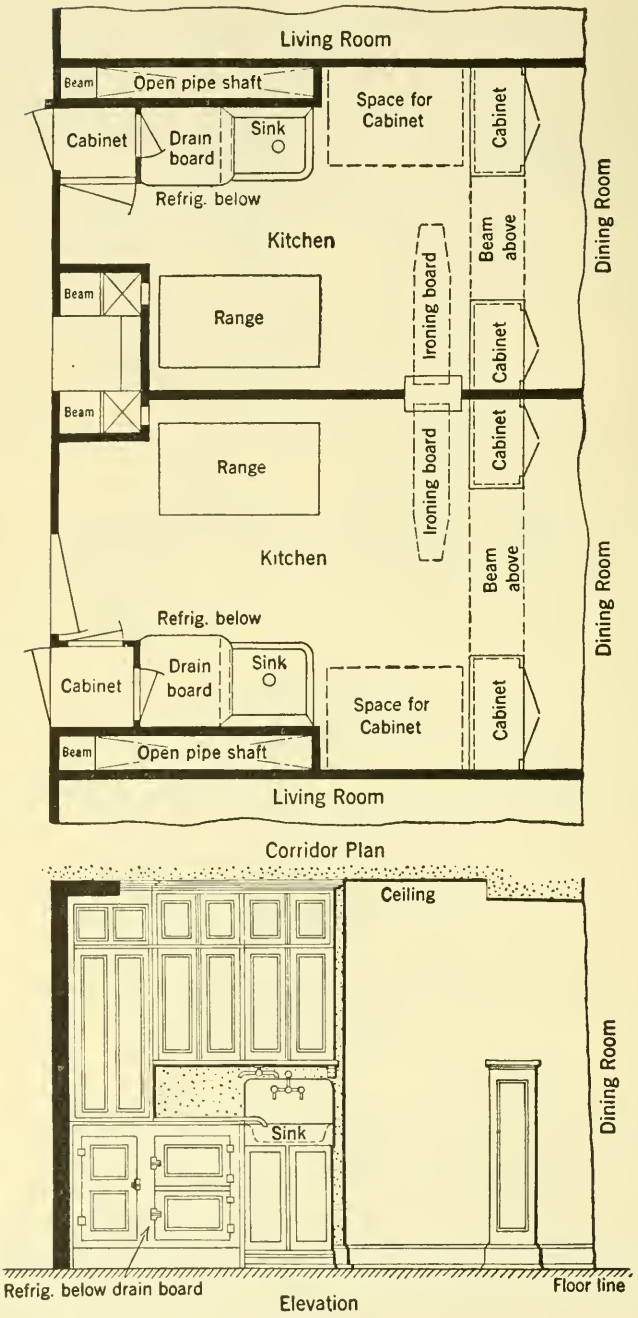



FIG. 360.—Typical Arrangement of Apartment.

than 3.0 deg. F. rise of temperature of the brine should be permitted during the circuit through the boxes.

The pumping head varies with each installation. With the balanced tank design the head is due to the friction head and the velocity head only. This may be calculated as in the problem on brine piping, Chapter VIII. Allowance for the fittings should always be made in calculating

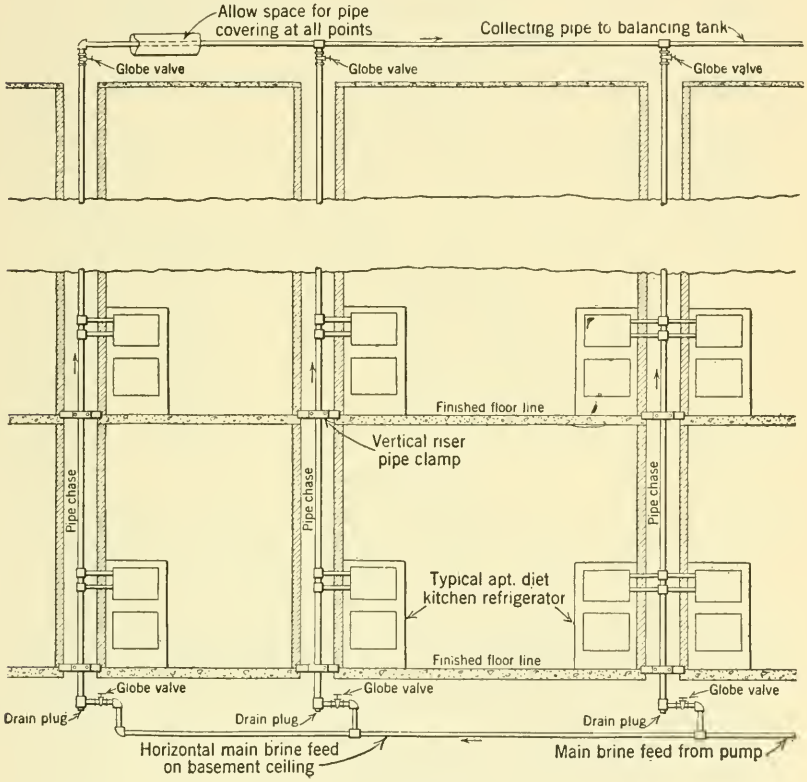



Fig. 361.—Apartment Piping.

the total head on the pump. The pressure on the brine tank or the piping at the pump is the " total head " plus the static head due to the elevation of the balance tank above the pump. Figures 350 to 361 give typical piping arrangements and typical apartment arrangment of boxes.

**The Drinking Water System.**—Factories and particularly office buildings and hotels are making use of the mechanically cooled drinking

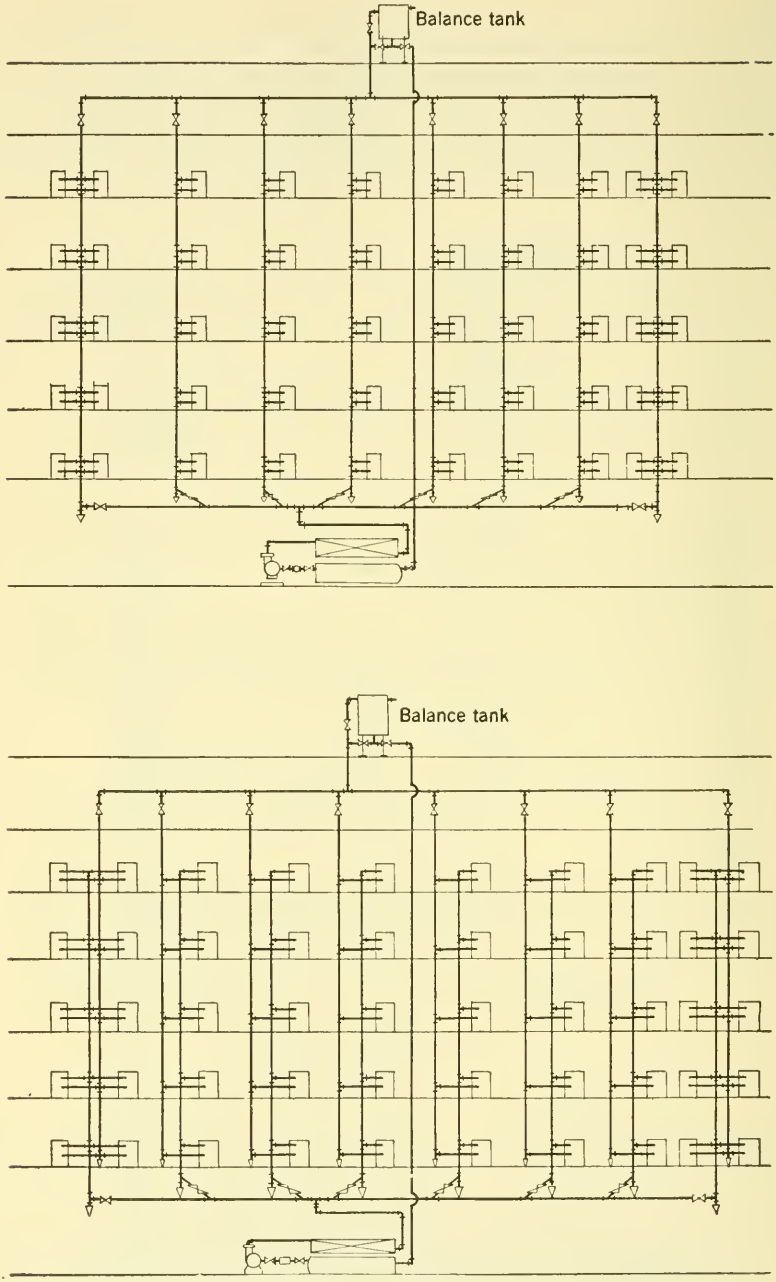



Fig. 362.—Brine Piping.

water system, as the use of the ice cooler is neither convenient, economical, nor hygienic. With drinking water systems the water may be

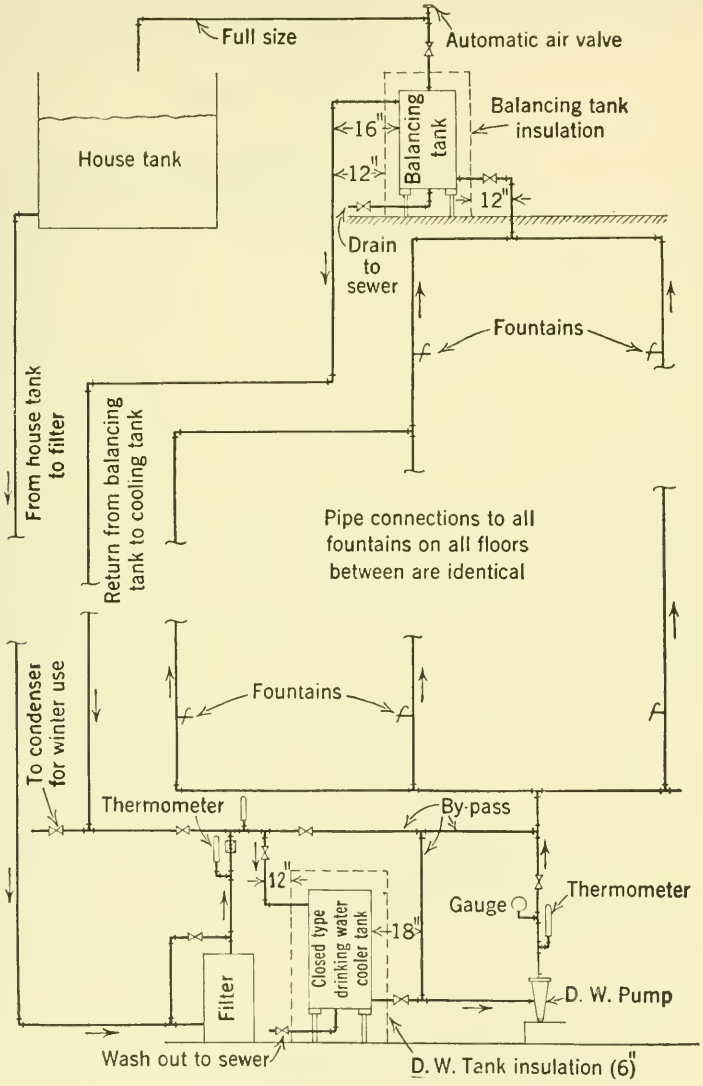



Fig. 363.—Drinking Water System.

filtered, is cooled to the desired temperature and, in particular, may be designed with convenient fountains for the use of the workmen or the office force in the different parts of the factory or office.

Such a refrigerating system consists of a water-cooling tank of one design or another, the refrigerating unit, a circulating pump to keep the water in constant circulation throughout the system, a balance tank and, finally, a make-up valve and float so as to replenish the water

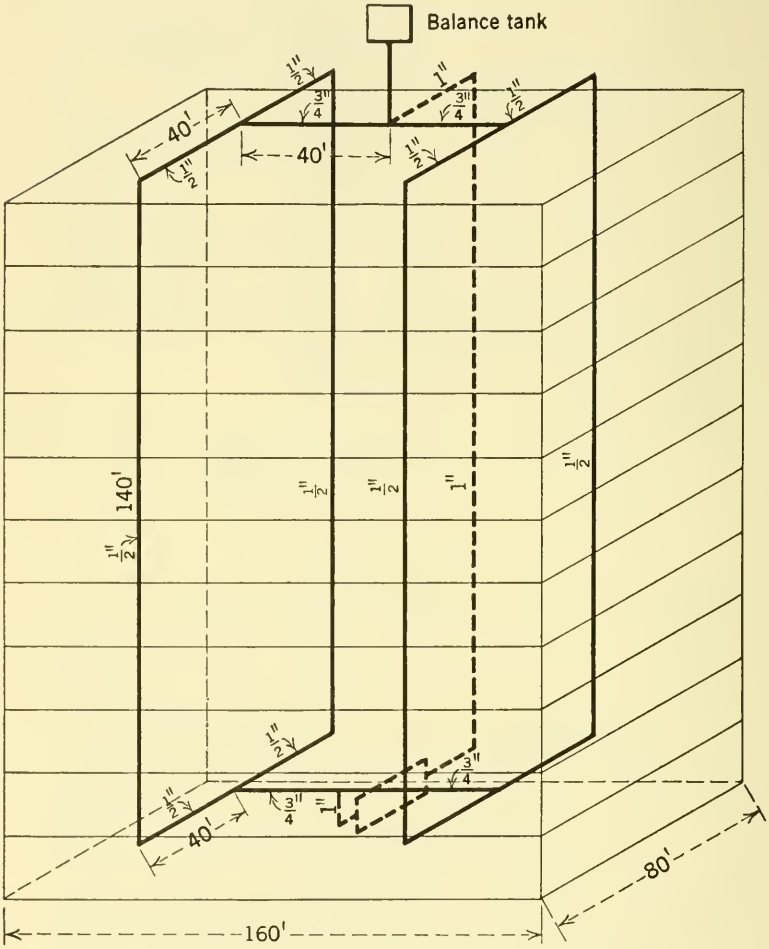



Fig. 364.—Problem in Drinking Water Systems.

supply as fast as it is used, sometimes in connection with the balance tank. Figure 363 shows a typical installation.

The water circuit must be complete at all times. By this is meant that from four to five times the water consumed, and wasted, must be

circulated in the pipes. This circulating water is required so as to have a positive pressure always at the outlets of all the fountains, and also so as to absorb and carry away the heat leakage through the pipe insulation. As the water temperature may be assumed to be 50 deg. F., and the outside air temperature 95 degrees, the heat leakage develops into large proportions at times.

The function of the refrigerating machine then is to cool the make-up water to, say, 50 degrees and to overcome the effect of heat leakage. The latter naturally depends on the kind of insulation used and the length and the size of the pipe in the water circuit. It has been found advantageous to use $\frac{3}{4}$ and 1-in. full weight pipe for the circuits as a rule, and to limit the circuits to 2000 ft., although 5000 ft. circuits have been successful. Too long a water circuit will represent an excessive pumping cost, and in addition to the increased power this work of pumping invariably will degenerate into a heating effect which in turn will have to be corrected by refrigeration at the rate of $\frac{1}{5}$ ton refrigeration per one horse power of the circulating pump. For the same reason the velocity of the water in the pipe should be limited to not more than 200 ft. per minute.

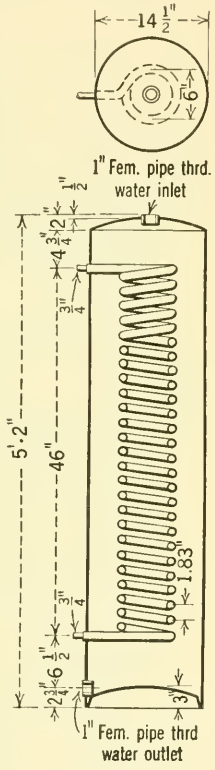



Fig. 365.—Drinking Water Cooler.

The amount of water *used and wasted* is a difficult variable to account for. It naturally is dependent on the severity of the labor of the workmen. Hotels and office buildings having an outlet in each room should be designed on the basis of *one gallon* per person per day. Figure 368 shows the amount of water consumed in one of the St. Louis hotels. This plant was laid out to take care of a demand of 1000 gal. per hour or 1.5 gal per faucet per hour. In all likelihood one-sixth of this amount would have been sufficient, or on the basis of $\frac{1}{4}$ gal. per faucet per hour. So it would seem that the rule of one gallon per person per day of eight hours is very liberal, although it will always be safe. Stores and office buildings providing outlets in isolated places in aisles or corridors may have these in use continuously, and they should be figured on the basis of a *half pint* per minute per fountain. In factories it is good practice to use *one-half gallon* per person per hour, and in offices *one-quarter gallon* per person per hour. These figures are much higher than those used in

the Federal Buildings,[1] which are designed with the uniform rule of *one pint* of water per person per hour irrespective of the kind of service the building may be called upon for.

The number of fountains are also variable, depending as this does on the kind of installation. The attempt is made in their location to

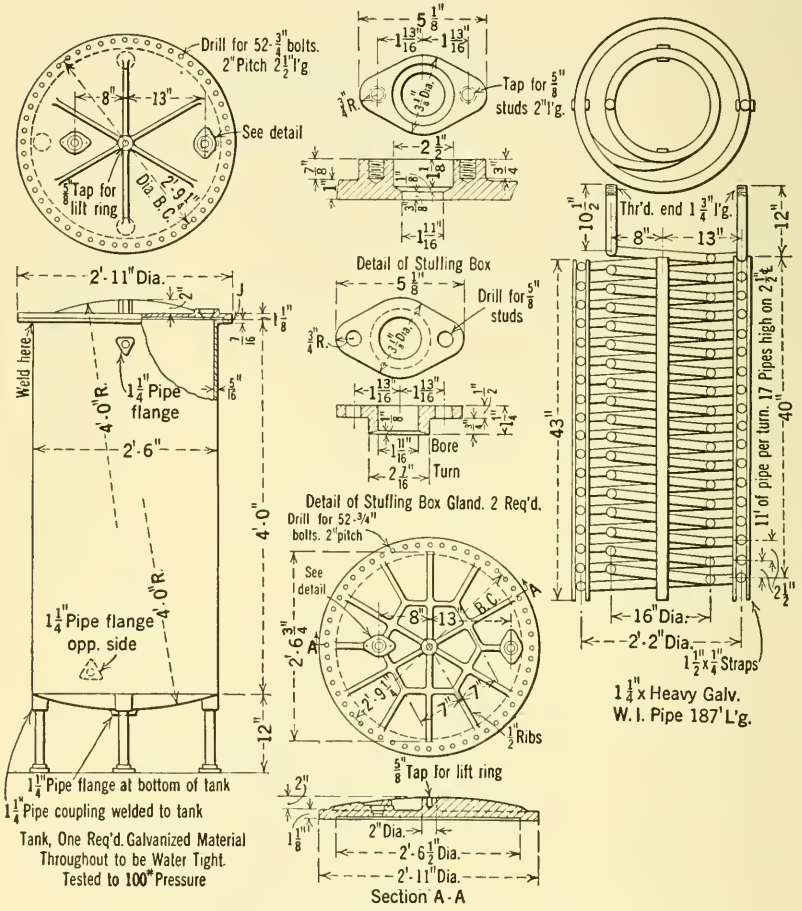

Drill for 52-3/4" bolts. 2" Pitch 2 1/2" l'g
See detail
Tap for lift ring
Dia. B.C.
Tap for 5/8" studs 2" l'g.
Detail of Stuffing Box
Drill for 5/8" studs
Bore
Turn
Detail of Stuffing Box Gland. 2 Req'd.
Drill for 52-3/4" bolts. 2" pitch
Thr'd. end 1 3/4" l'g.
Weld here
1 1/4" Pipe flange
1 1/4" Pipe flange opp. side
1 1/4" Pipe flange at bottom of tank
1 1/4" Pipe coupling welded to tank
Tank, One Req'd. Galvanized Material Throughout to be Water Tight. Tested to 100* Pressure
1/2" Ribs
5/8" Tap for lift ring
Section A-A
11' of pipe per turn. 17 Pipes high on 2 1/2" ℄
16" Dia.
2'-2" Dia.
1 1/2" x 1" Straps
1 1/4" x Heavy Galv. W. I. Pipe 187' L'g.

Fig. 366.—Drinking Water Cooled by Means of Carbon Dioxide.

occasion as little difficulty as possible for each workman, and to work on the assumption that the first cost of extra fountains is preferable to the cost of the lost time of the workman in walking to a distant fountain. In consequence, good practice will allow one fountain per 100 operators

[1] Nelson S. Thompson, Supervising Architects Office, Treasury Department.

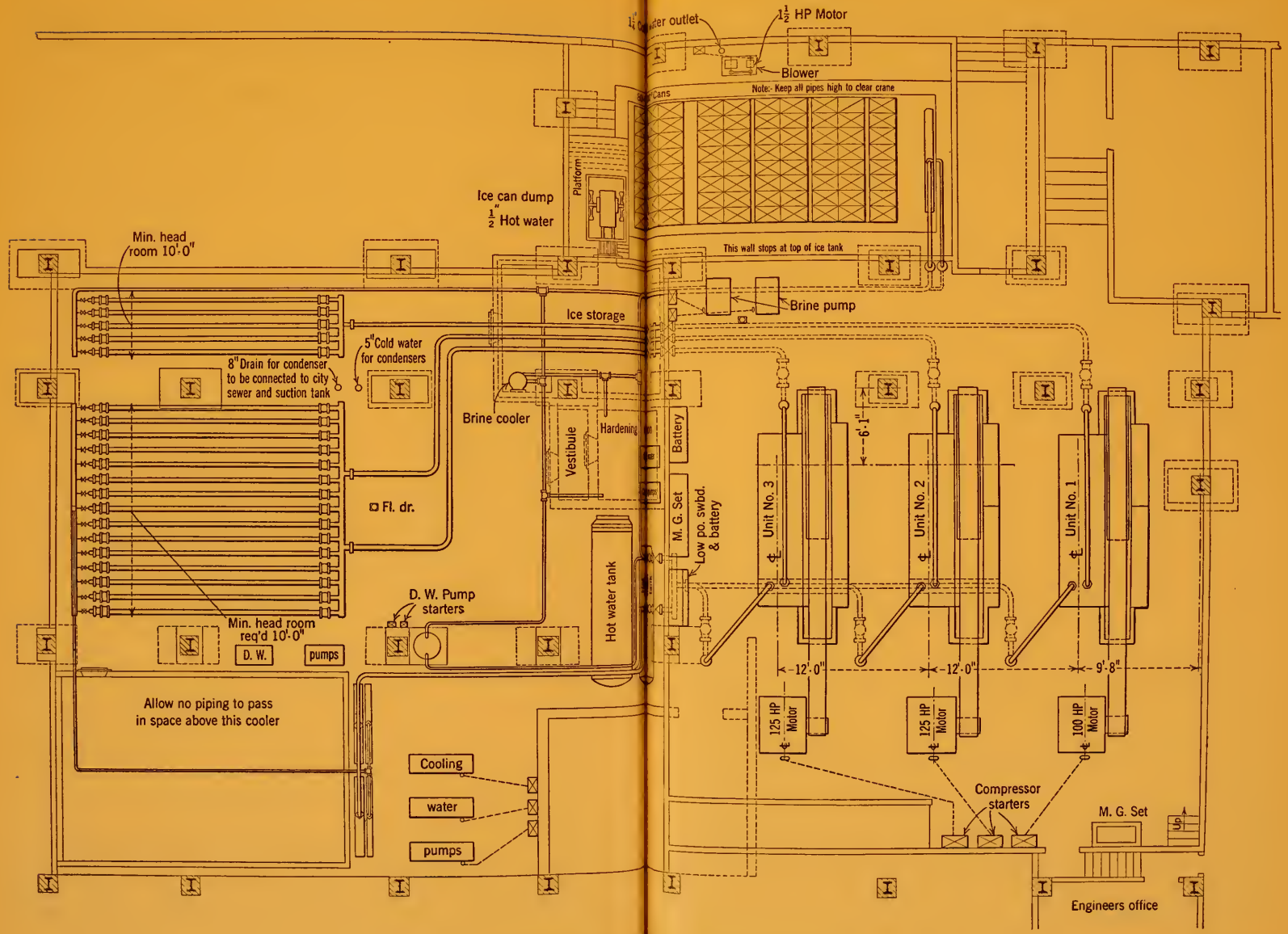



FIG. 367.—Hotel Refrigeration, using Carbon Dioxide.

*To face page 579.*

or per 50 workmen, depending on the details of the job and the congestion of the shop or office.

The kind of insulation used is also very important from the viewpoint of the heat leakage, and Table 116 can be used advantageously for ice water conditions, the corkboard and the lithboard being $1\frac{1}{2}$ in. thick and the hair felt being erected in two layers. The heat transfer values are for *one foot* of length per *one degree* difference in temperature *per hour*.

TABLE 116

Coefficient of Heat Transfer, per Linear Foot

| | Union Lith | Hair Felt (Two Layers) | Corkboard |
|---|---|---|---|
| $\frac{1}{2}$-in. pipe | 0.157 | 0.086 | 0.160 |
| $\frac{3}{4}$-in. pipe | 0.170 | 0.095 | 0.167 |
| 1 -in. pipe | 0.180 | 0.107 | 0.178 |
| $1\frac{1}{4}$-in. pipe | 0.213 | 0.124 | 0.199 |
| $1\frac{1}{2}$-in. pipe | 0.217 | 0.132 | 0.220 |
| 2 -in. pipe | 0.269 | 0.149 | 0.245 |
| $2\frac{1}{2}$-in. pipe | 0.272 | 0.167 | 0.291 |
| 3 -in. pipe | 0.294 | 0.192 | 0.304 |
| $3\frac{1}{2}$-in. pipe | 0.355 | 0.211 | 0.330 |
| 4 -in. pipe | 0.367 | 0.228 | 0.345 |
| $4\frac{1}{2}$-in. pipe | 0.376 | 0.247 | 0.385 |
| 5 -in. pipe | 0.466 | 0.268 | 0.410 |
| 6 -in. pipe | 0.558 | 0.307 | 0.437 |

**Problem.**—The following problem illustrates the manner of calculation of the size of a drinking water system. An office building 160 by 80 ft. and 10 stories high is to have a drinking water system with the water to be delivered at a temperature of 50 deg. F. The floors are 13 ft. high, the make-up water is at a temperature of 80 degrees and 3.0 gal. are to be provided per person per day of 8 hours.

The floor area is 12,800 sq. ft. and, with an allowance of one person per 70 sq. ft.[2] of floor area, there will be $\frac{12,800}{70} = 183$ persons per floor. Using one fountain per 50 persons there will be required 4 fountains per floor and 40 fountains in all. The piping will be arranged as shown in the figure, which has four circuits in parallel. Assume that the risers will be of $\frac{1}{2}$-in. pipe using 225 ft. of $\frac{1}{2}$-in. pipe, 80 ft. of $\frac{3}{4}$-in. pipe, and 220 ft. of 1-in. pipe in each of the circuits, except that the 1-in. pipe is the common return. The heat absorbed using an average temperature of the water of 47.5 degrees, 95 degrees outside temperature, and corkboard covering, is as follows:

[2] The factory allows one person per 100 sq. ft.

$\frac{1}{2}$-*in. line.* $Q = 225 \times 0.160 \times (95 - 47.5) = 1710$ B.t.u. per hour (per riser)

$= 6840$ B.t.u. per hour.

$\frac{3}{4}$-*in. line.* $Q = 160 \times 0.167 \times (95 - 47.5) = 1270$ B.t.u.

1-*in. line.* $Q = 220 \times 0.178 \times (95 - 47.5) = 1860$ B.t.u.

Total $= 9970$ B.t.u. per hour.

The amount of water required to absorb this heat, assuming a rise of temperature of the water of $4\frac{1}{2}$ degrees,

$$\frac{9970}{4.5} = 2216 \text{ lb. per hour}$$

$= 4.43$ gal. per min. total
$= 1.11$ gal. per min. per circuit.

*Make-up water.* $3.0 \times 10 \times 183 = 5490$ gal. per 8 hr.
$= 686$ gal. per hr. $= 11.45$ gal. per min.
$= 2.86$ gal. per min. per riser.

Total water required, neglecting the effect of friction heating = 15.88 gal. per min.

*Refrigeration required.*

$$8.33 \times 60 \times 11.45 \times (80 - 45) + 9970 = 210{,}400 \text{ B.t.u.} = 17.6 \text{ tons.}$$

The friction developed in the water circuit is (Fig. 180):

$\frac{1}{2}$-in. line (3.97 gal. per min.) = 2.6 in. of water per ft.
$\frac{3}{4}$-in. line (7.94 gal. per min.) = 2.3 in. of water per ft.
1-in. line (15.88 gal. per min.) = 2.3 in. of water per ft.

Total friction $= (2.6 \times 225) + (2.3 \times 80) + (2.3 \times 220) = 1275$ in. of water
$= 46.0$ lb. per sq. in.

The heating effect of the friction head is

$$15.88 \times 8.33 \times 46.0 \times 2.31 = 14{,}050 \text{ ft.-lb. per min.}$$

$$\frac{14{,}050}{33{,}000} \times 42.4 = 18.1 \text{ B.t.u. per min.}$$

The size of the motor will be $0.425 \div 0.5 = 0.850 = 1.00$ hp.

Figure 365 gives details of a small water cooler and Fig. 366 gives the construction of a typical water cooler using carbon dioxide as a refrigerant. The carbon dioxide under these circumstances is always *placed in the pipes.* Figure 367 gives a representative layout for carbonic refrigeration in a hotel. Figure 368 shows graphically the demand on a drinking water system in the Statler Hotel, St. Louis, for a single day and for a five-day period. This system was designed for a maximum demand of 1000 gal. per hour.

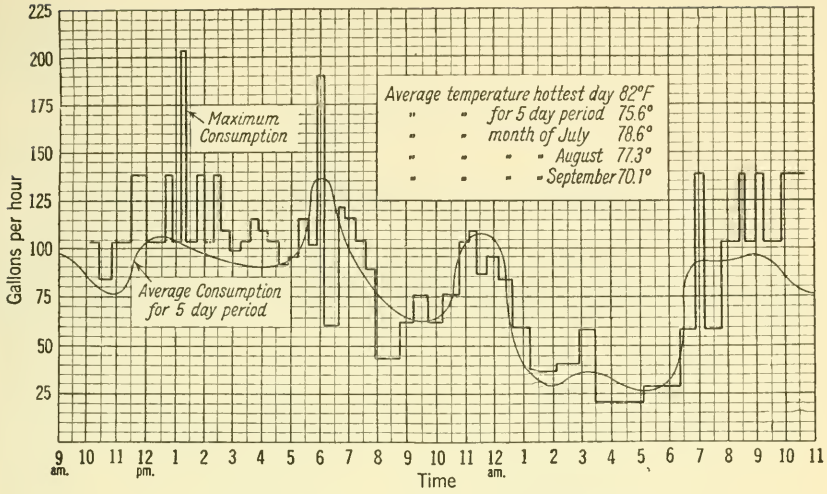



FIG. 368.—Load Curve for Hotel Drinking Water.

TABLE 117

PIPING RATIO FOR SMALL REFRIGERATORS—NUMBER OF CUBIC FEET OF SPACE COOLED BY ONE RUNNING FOOT OF BRINE COIL

| For Refrigeration up to 200 Cubic Feet | | For Refrigeration up to 500 Cubic Feet | | For Refrigeration up to 1000 Cubic Feet | | For Refrigeration up to 2000 Cubic Feet | |
|---|---|---|---|---|---|---|---|
| Temperature, degrees F. | Cubic feet | Temperature, degrees F. | Cubic feet | Temperature, degrees F. | Cubic feet | Temperature, degrees F. | Cubic feet |
| 40 | 3.0 | 40 | 4.0 | 40 | 8.0 | 40 | 10.0 |
| 36 | 2.0 | 36 | 3.0 | 36 | 6.25 | 36 | 8.0 |
| 32 | 1.75 | 32 | 2.5 | 32 | 5.0 | 32 | 6.5 |
| 20 | 1.25 | 20 | 1.75 | 20 | 3.75 | 20 | 5.25 |
| 10 | .7 | 10 | 1.25 | 10 | 2.50 | 10 | 4.0 |
| 5 | .5 | 5 | .75 | 5 | 1.25 | 5 | 2.25 |
| 0 | .25 | 0 | .35 | 0 | .625 | 0 | 1.25 |

SIZES OF BRINE MAINS FOR SMALL REFRIGERATORS

$\frac{1}{2}$-in. Feed and return for $1\frac{1}{4}$-in. Coils up to 200 lineal feet.
$\frac{3}{4}$-in. Feed and return for $1\frac{1}{4}$-in. Coils up to 400 lineal feet.
1 -in. Feed and return for $1\frac{1}{4}$-in. Coils up to 800 lineal feet.
$1\frac{1}{4}$-in. Feed and return for $1\frac{1}{4}$-in. Coils up to 1500 lineal feet.

## CHAPTER XVIII

## REFRIGERATION IN THE CHEMICAL INDUSTRIES

There is no question but that mechanical refrigeration will play an increasingly important part in the development of processes in the chemical industry. Mechanical cooling may be used to hasten the process, but at times it is absolutely necessary not only to condition the air or to secure moderately cold temperatures, but to obtain very low temperatures. As a rule the actual cooling problem is relatively simple, as it is usually simply the cooling of a liquid, or the cooling and freezing of a liquid, and the constants involved in the calculation of this process can be secured only by experiment. A large number of the chemical processes are secret, but a few of the more generally known applications will be given in this chapter with the understanding that other processes are similar in their method of solution.

**Low Temperatures.**—In America low temperatures have been secured by the use, first, of the absorption machine and, more lately, by the use of the stage compressor. The Eastman Kodak Company, which uses in their process work a − 20 deg. F. brine, has three 700-ton two-stage ammonia compressors. In Great Britain low temperature refrigeration has been obtained by means of the carbonic compressor which has been developed to a much higher degree than it has in the United States. For − 60 deg. F. refrigeration the two-stage carbonic compressor, built by the Liverpool Refrigerating Company, has had outstanding success. The pressure in the evaporator for − 60 deg. F. is about 100 lb. per sq. in. with carbon dioxide, whereas with ammonia it would be only 5.5 lb. absolute.

**Cooling Corrosive Substances.**—Corrosive substances can be cooled best by the use of brine. This brine piping can be designed for slight unit pressures and the piping can be made of lead, porcelain covered, or even of glass.

**Removal of Salts from Solution.**—Certain salts may be extracted best by means of crystallization and precipitation incidental to a lowering of the temperature. For example, a saturated solution of sodium nitrate, $NaNO_3$, at 90 deg. C., will hold in solution 162 grams of salt in 100 cc. of water but at 20 deg. C. the amount will be reduced to 85 grams and

the difference (77 grams) must be precipitated out of solution. The following problem is a good example of this sort of application of refrigeration.

**Problem.**—10,000 lb. of water saturated with potassium chlorate, $KClO_3$, is to be cooled from 68 deg. F. to 14 deg. F. per hour. Find the refrigeration and the piping required for the heat transfer.

One arrangement of the apparatus required is shown in (Fig. 369). This apparatus consists of two cylindrical tanks with conical bottoms, the latter fitted with cocks of ample size for quick operation. Referring to the figure it will be seen that the strong solution enters the left-hand tank at 68 deg. F. and passes downward, being cooled on the way, with precipitation of some potassium chlorate

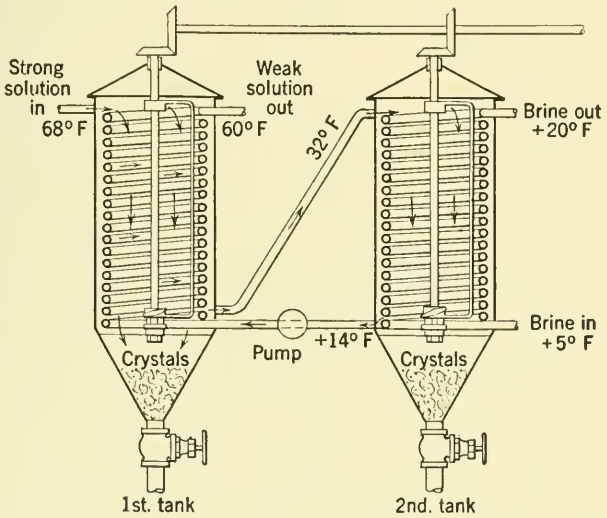



Fig. 369.—Removal of Salts from Solution.

crystals, and finally passes over into the right-hand tank by means of the difference in liquid level or by means of a pump. In the second tank is arranged a helical coil for brine cooling, the brine entering at 5 deg. and leaving at 20 deg. F. The solution is thereby lowered in temperature to 14 degrees and leaves the tank at this temperature to be pumped through a helical coil in the first tank where it leaves at 60 deg. F. The first tank (the one on the left) is therefore simply a heat exchanger, intended to give economy of action by the utilization of all the refrigeration existing in the liquid after the process that it is practical to attempt to use.

As the crystals will tend to form on the cooling coils it is necessary to provide some means of continually scraping them, and this is done by the wire brushes attached to the vertical shafts, as shown in the figure. The crystals settle into the conical part of the tank and are drawn off into other tanks, the upper part of which is separated from the lower by means of wire screen so that what liquid passes along with the solid potassium chlorate will pass through the screen.

To precipitate the potassium chlorate there will be required 146.7 B.t.u. of refrigeration per pound of salt formed. Referring to the chart (Fig. 370), (7.4 − 2.2) =

5.2 grams of salt is precipitated per 100 cc. of water, or 5.2 per cent of the weight of the water is the weight of the salt removed from solution, therefore 520 lb. of salt crystals will be produced per hour. During the process in the first tank the weak solution will be heated from 14 to 60 deg. F., and the heat absorbed, taking the specific heat of the solution at 0.85 will be:

$$Q = 10{,}220 \times 0.85 \times (60 - 14) = 400{,}000 \text{ B.t.u.},$$

and the strong solution will lose an equal amount of heat as heat of the liquid and as heat of the solution. By trial and error it works out that, by allowing for losses through the insulation the temperature of the strong solution leaving the first tank will be 30 degrees, and that the strong solution will be reduced to a concentration of 3.3 grams per 100 cc. of water whereby 410 lb. of salt will be precipitated and the remaining 110 lb. of chlorate will be deposited in the tank with the brine coil.

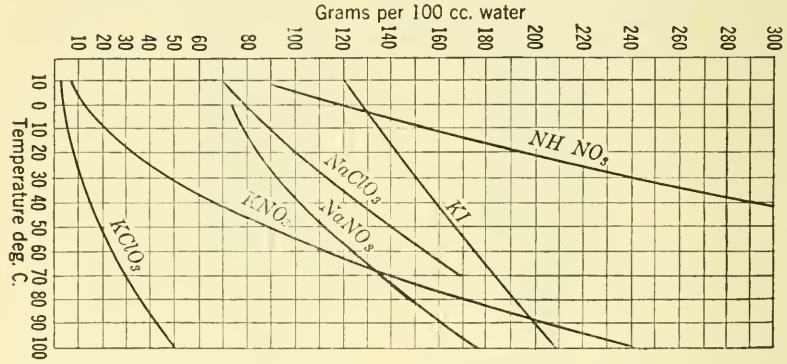



Fig. 370.—Solubility of Salts at Various Temperatures.

The load on the brine coil will be (taking the heat of precipitation of the salt solution as 146.7 B.t.u.),

$$Q = (10{,}270 \times 0.85 \times 30 - 14) + (110 \times 146.7) = 155{,}800 \text{ B.t.u.}$$

$$= 12.99 \text{ tons of refrigeration.}$$

To this 13.0 tons of refrigeration must be added 5 per cent for losses through the insulation, which makes the total load on the refrigerating machine 13.64 tons. The compressor displacement, for 15 lb. suction and 160 condenser pressure, will be $13.64 \times 4.7 = 64.1$ cu. ft. per minute, and the power required to drive the compressor will be $1.42 \times 13.64 = 19.37$ hp., but a larger motor will be required for starting and as a factor of safety. For this size of compressor the best design is the single-acting vertical enclosed type of machine which will operate at about 200 r.p.m. Using these values, the size of compressor works out to be 7.1 by 7.1, so that the 7 by 7 or the $7\frac{1}{2}$ by $7\frac{1}{2}$ standard sizes would be used, making the proper adjustment of speed to suit.

The amount of pipe surface for the coils in the tanks may be considered from the following viewpoint. The inside of the coils will be kept free of crystals because of

the action of the rotating brushes, and the propeller will keep the liquid agitated. A value of 50 to 75 for the coefficient of heat transfer is justified if all the surface is effective, but as half is coated with salt it will be safer to use a value of 40. The mean temperature difference in the first tank (the logarithmic mean) is 12.35 (Fig. 163 and 371), and the surface becomes:

$$400{,}000 = A \times 40 \times 12.35$$

$$A = 810 \text{ sq. ft.} = 1295 \text{ ft. of 2-in. pipe.}$$

This surface could be provided in a coil 8 ft. in diameter with 52 turns. If the pipe coil has a pitch of 3 in. the coil would be 13 ft. high, with $\frac{5}{8}$ in. between each pipe. In a similar manner the pipe coil required for the second tank would be 410 sq. ft., or 656 lin. ft. of 2-in. pipe. The two tanks should be covered with 4 in. of cork or the equivalent.

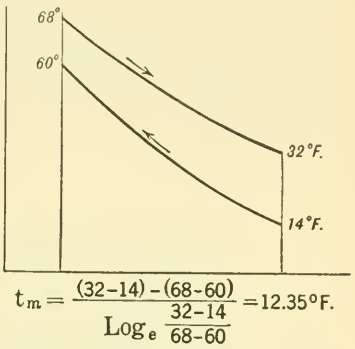



FIG. 371.—Problem. Method of Securing the Mean Temperature Difference.

**Oil Refining.**—The oil refineries manufacturing a low "cold test" oil use a procedure almost similar to the process just described except for the details of the plant. The problem again involves the cooling of a liquid and the congealing of a part of this liquid.

*The Wax Process.*—In the oil refining process, in order to get the required "cold" test for certain grades of lubricating oils it is necessary to chill the paraffine wax distillate and separate the resulting congealed paraffine from the oil by pumping through a filter press. The problem involves the cooling of brine, as direct expansion is seldom used, and a suitable device for chilling the oil. As the wax congeals, some scraping attachment must be included in order that as the solid material collects on the pipe surface it may be scraped away and pushed along in the direction of flow of the chilled oil. The usual design is to use a double pipe cooler with a screw conveyor in the inner pipe (Fig. 372) so constructed as to give a positive flow of the oil and wax and an efficient clearing of the oil surface at all times. It is very necessary to have a clean surface of the piping if results of value are to be obtained.

The amount of wax and the physical constants in distillate vary with the different fields, but the average is taken as 0.5 for the specific heat of the oil, and 0.87 for the specific gravity. The latent heat of fusion of the paraffine is 125 B.t.u. per lb. and it begins to separate out at 60 deg. F. As an illustrative example, 1000 gal. of distillate per hour are to be cooled from 80 to 20 deg. F. and 10 per cent of wax by weight will be removed. Assume the specific heat of the solid wax as 0.6. Find the size of the compressor required, allowing 10 per cent for losses by heat

leakage, etc., and find the size of the double pipe cooler required for cooling the oil.

The weight of oil per hour is $1000 \times 8.33 \times 0.87 = 7{,}250$ lb.

| | | B.t.u. |
|---|---|---|
| *a*. To cool the oil to 60 deg. F. | $0.5 \times 7250 \times (80 - 60) =$ | 72,500 |
| *b*. To freeze the paraffine | $0.1 \times 7250 \times 125 =$ | 90,600 |
| *c*. To cool the oil | $0.9 \times 0.5 \times 7250 \times (60 - 20) =$ | 130,500 |
| *d*. To cool the wax | $0.1 \times 0.6 \times 7250 \times (60 - 20) =$ | 17,400 |
| Total | | 311,000 |
| Add 10 per cent for safety | | 31,100 |
| Total [1] | | 342,100 |
| | | 28.5 tons. |

The resulting clarified oil, being at 20 degrees, could be used to decrease the refrigerating load by pumping it back to a counterflow double pipe cooler (and permit it to warm up to 40 deg. F. or more), through which the distillate at 80 degrees is passed. In fact the oil cooler is designed at times using the two uppermost pipes as such an "exchanger." If such a device were used, the refrigeration saved could be expected to be:

$$0.9 \times 7250 \times (40 - 20) \times 0.5 = 65{,}250 \text{ B.t.u.},$$

and the tonnage becomes

$$\frac{1.1 \times 245{,}750}{12{,}000} = 22.5 \text{ tons of refrigeration.}$$

But the oil cooler would be less efficient in heat transfer because of having oil on both sides of the inner pipe and also because of the lesser temperature difference when using oil and brine, as compared with brine throughout the cooler, for the two uppermost pipes. As a rule zero degree brine is used in these coolers, and for some time the absorption machine has been the favorite means of securing refrigeration because exhaust steam is always available in refineries and the absorption machine can use this steam directly in the still to generate the high pressure ammonia gas required for liquefaction in the condenser with ordinary condenser water.

The distillate chilling machine, Fig. 372, is made of 4-in. and 2-in. pipes as shown, the brine being in the outer and the distillate in the inner of the pipes. The amount of the cooling surface is more or less empirical as, for example, one refinery uses 400 lin. ft. of cooling pipe surface to

[1] For general design purposes it is usual to follow standard practice of about 7 to 8 barrels of wax distillate per hour per 1 ton of refrigeration.

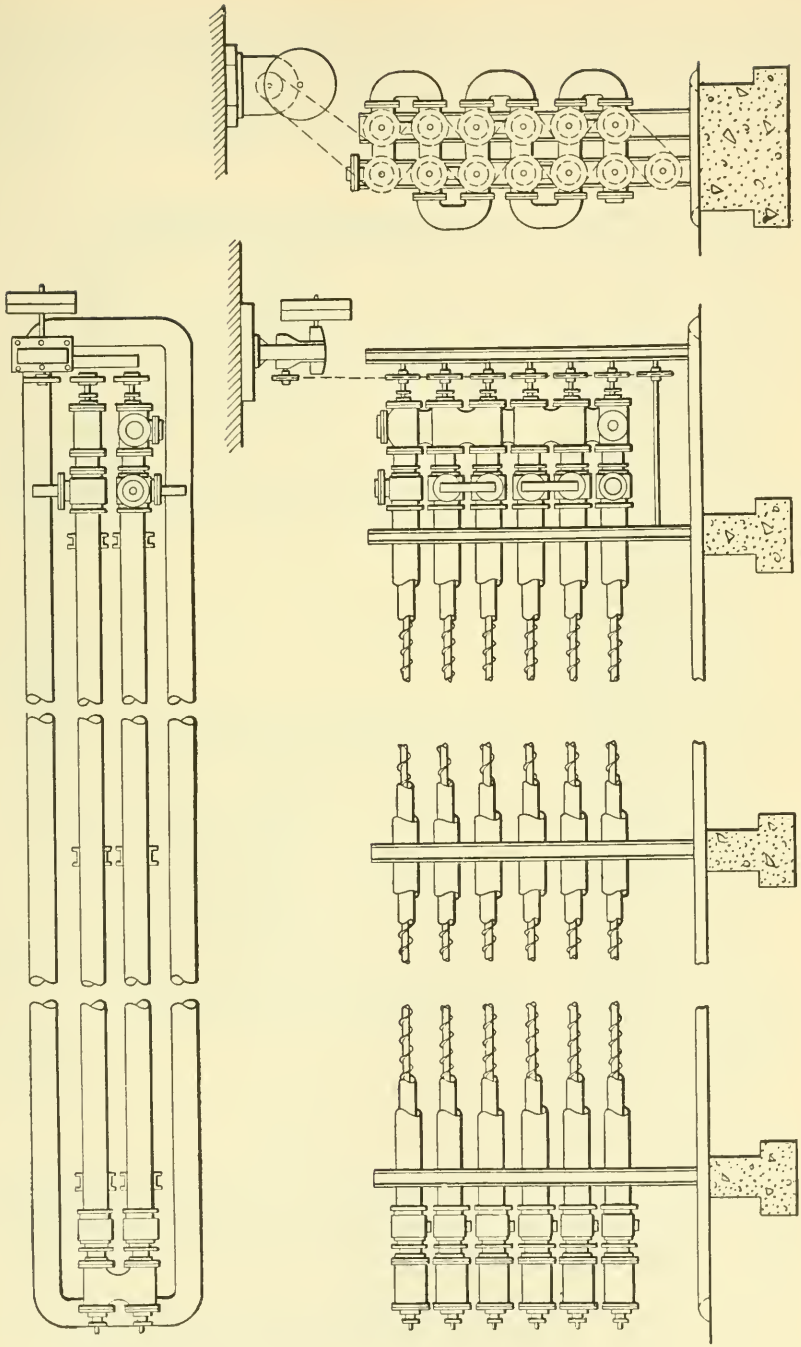


FIG. 372.—The Wax Chilling Machine.

chill 20 gal. of distillate per minute from 80 to 20 degrees using zero degree brine. In this problem the mean temperature difference is

$$t_m = \frac{(80 - 5) - (20 - 0)}{\log_e \frac{80 - 5}{20 - 0}} = 41.6 \text{ degrees}$$

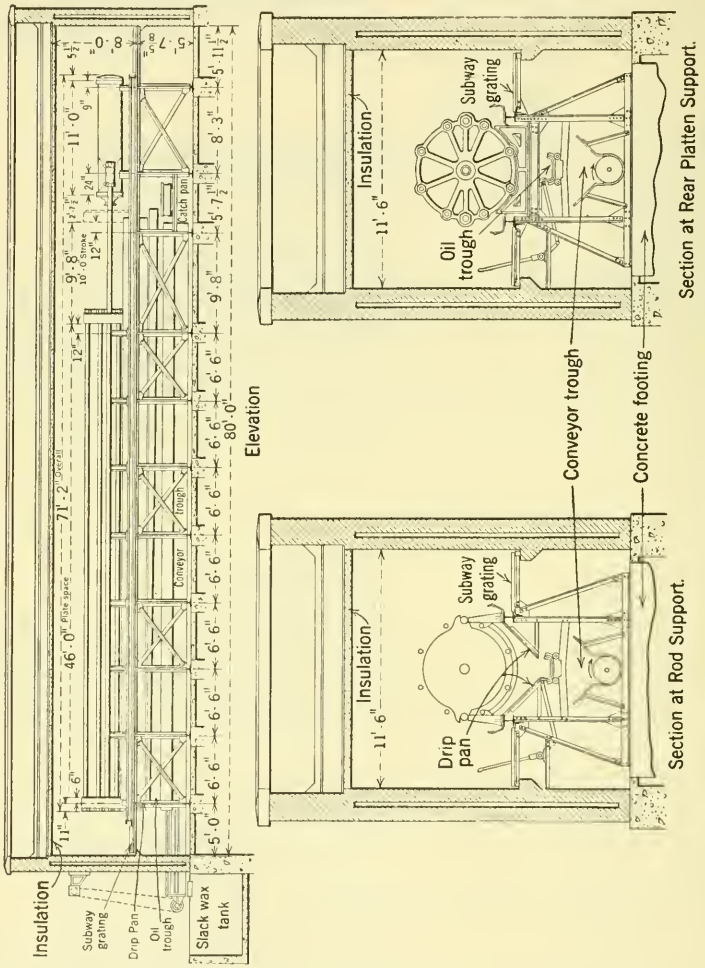



Fig. 373.—Filter Press for Wax Separation.

with a rise of 5 degrees in the temperature of the brine from 0 to 5 deg. F. Using a value of $k$ of 25, the required surface becomes

$$311{,}000 = A \times 25 \times 41.6.$$

$$A = 298 \text{ sq. ft.}$$

$$= 298 \times 1.6 = 478 \text{ linear feet of 2-in. pipe.}$$

The value of 25 for $k$ is possible only when the surface is kept carefully scraped of the wax, which means that the screw conveyor must make a good fit in the inner pipe. Figures 372 and 373 give details of the process. Table 118 contains the constants for the more general sub-

TABLE 118

THERMAL PROPERTIES OF VARIOUS SUBSTANCES

| Substance | Temperature, Deg. F. | Specific Gravity | Specific Heat | Latent Heat, B.t.u. per Lb. | Temperature, Deg. F. | | |
|---|---|---|---|---|---|---|---|
| | | | | | Freezing | Melting | Boiling |
| Alcohol, ethyl ($C_2H_6O$) | 32 | ..... | 0.547 | | | | |
| | 104 | 0.794 | 0.648 | ..... | −180.0 | ..... | 173.0 |
| | 176 | ..... | 0.769 | | | | |
| Alcohol, methyl, $CH_4O$ | ..... | ..... | 0.60 | | | | |
| Beeswax, yellow | ..... | 0.960 | 0.45 | 50.0 | 144.0 | 144.0 | |
| Beeswax, white | ..... | 0.965 | ..... | ..... | 146.0 | 146.0 | |
| Beef tallow, fresh | ..... | 0.968 | ..... | ..... | 91.4 | 109.4 | |
| Beef tallow, old | ..... | ..... | ..... | ..... | 93.0 | 110.5 | |
| Glycerine | ..... | 1.26 | 0.576 | ..... | ..... | ..... | 554.0 |
| Mutton tallow, fresh | ..... | 0.92 | ..... | ..... | 97.0 | 116.6 | |
| Benzol, $C_6H_6$ | 22 | ..... | 0.313 | | | | |
| | [Solid] | | | | | | |
| | 59 | 0.885 | 0.340 | 54.2 | | | |
| | 104 | ..... | 0.423 | ..... | ..... | 42.0 | 176.0 |
| | 149 | ..... | 0.482 | | | | |
| Butter, fresh | 212 | 0.866 | 0.55 | ..... | 66 to 68 | 88.0 | |
| Cocoa butter | ..... | 0.89 to | | | | | |
| | | 0.91 | ..... | ..... | 69.0 | 93.0 | |
| Cocoanut oil | ..... | ..... | ..... | ..... | 68.5 | 76.0 | |
| Nutmeg butter | ..... | ..... | ..... | ..... | 91.4 | 111.0 | |
| Glue | ..... | 1.27 | | | | | |
| Hexane [$C_6H_{14}$] | 32 | 0.658 | | | | | |
| Leaf lard | ..... | 0.92 to | | | | | |
| | | 0.94 | 0.60 | 90.0 | ..... | ..... | 156.0 |
| Milk, cows | ..... | 1.028 to | | | | | |
| | | 1.035 | | | | | |
| Naphtha | ..... | 0.85 | 0.45 | | | | |
| Naphthalin, solid, $C_{10}H_8$ | 50 to 68 | 1.152 | 0.314 | | | | |
| | 140 | ..... | 0.334 | 64.8 | | | |
| Naphthalin, liquid, $C_{10}H_8$ | 200 | 0.977 | 0.41 | | | | |
| Nitric acid | ..... | 0.661 | | | | | |
| Oil, petroleum | 100 to | 0.79 to | | | | | 392 to |
| | 200 | 0.82 | 0.50 | ..... | ..... | ..... | 482 |
| | ..... | 0.88 | 0.51 | | | | |
| Oil, crude, Pa | ..... | 0.81 | 0.50 | | | | |
| Oil, crude, Japan | ..... | 0.86 | 0.45 | | | | |
| Oil, crude, Russian | ..... | 0.91 | 0.43 | | | | |
| Oil, crude, California | ..... | 0.96 | 0.40 | | | | |
| Oil, liquid base | ..... | 0.83 to | | | | | |
| | | 0.87 | 0.40 | | | | |
| Oil, lubricating | ..... | 0.83 to | | | | | 590 to |
| | | 0.87 | 0.40 | ..... | ..... | ..... | 660 |
| Oil, cottonseed | ..... | ..... | 0.47 | | | | |
| Oil, olive | 44 | 0.91 | 0.47 | | | | |
| Oil, linseed | ..... | 0.93 to | | | | | |
| | | 0.94 | 0.31 | ..... | −4.0 | ..... | 600 |
| Oil, turpentine, $C_{10}H_{16}$ | −4.0 | ..... | 0.38 | | | | |
| | 32.0 | ..... | 0.41 | | | | |
| | 176 | 0.87 | 0.48 | | | | |
| | 320 | ..... | 0.51 | ..... | ..... | ..... | 320.0 |
| Oil, castor | ..... | ..... | 0.43 | | | | |
| Oil, palm, soft | ..... | 0.905 | ..... | ..... | 69.8 | 86.0 | |
| Oil, palm, hard | ..... | ..... | ..... | ..... | 75.2 | 100.4 | |
| Oil, palm (old) | ..... | ..... | ..... | ..... | 100.4 | 107.6 | |
| Paraffin, soft | ..... | 0.87 to | | | | 100 to | 660 to |
| | | 0.88 | 0.70 | 63.0 | ..... | 125 | 730 |
| Paraffin, hard | ..... | 0.88 to | | | | | |
| | | 0.93 | ..... | ..... | ..... | 130.0 | 750.0 |
| Spermaceti | ..... | 0.88 to | | | | | |
| | | 0.94 | ..... | 46.4 | ..... | 112.0 | |
| Water, sea | ..... | 1.026 | 0.938 | ..... | 27.5 | | |

stances. Unfortunately, these values vary considerably unless the substance is a fixed compound.

*The Manufacture of Gasoline.*—There is no definite compound called gasoline, but the term is applied to a number of hydrocarbons including propane, butane, hexane, nonane, etc., including a little ethane, all mixed (or dissolved) together. The separation of these hydrocarbon compounds from gas by the means of refrigeration can be accomplished in the case of either casing head or natural gas. The freezing of the gasoline may be carried out by either of two methods. First, there

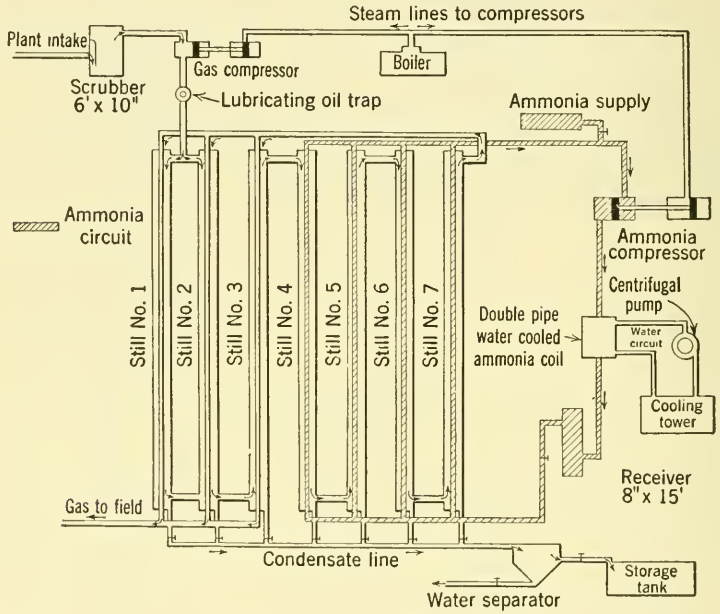



Fig. 374.—Gasoline Manufacture.

is the method by which the gas is compressed (to 160 lb. or over) and then is allowed to throttle down or expand adiabatically in the expansion cylinder of an engine. The temperature is lowered in both of these methods but to a much greater degree in the case of the expanding cylinder, and the gasoline fractions are condensed out during the fall of the temperature. A third method is when the gas is chilled by the use of a refrigerating machine and a suitable exchanger.

Raw casing head gas emanating from producing oil wells is pumped by vacuum pumps to a central point at which it is picked up by a compressor and compressed to a pressure varying from 5 to 40 lb. or more. After this operation the gas passes through a series of water cooled

atmospheric condensers, during which process the heavier gasoline fractions contained in the raw casing head gas are condensed and collected in an accumulator tower 30 to 90 ft. high filled with tile, wooden baffles, lath cuttings, or other special devices manufactured for the purpose of giving a large amount of surface for the absorbent oil to spread over in the downward course through the tower. The oil in the thin film state coming into contact with the upward passing gas absorbs the gasoline content of the gas. This oil passes from the bottom of the tower into the heat exchanger, thence to the pump, which forces it through some water cooled atmospheric coils, thence into a still where the gasoline content is driven off by heat. The denuded oil then passes

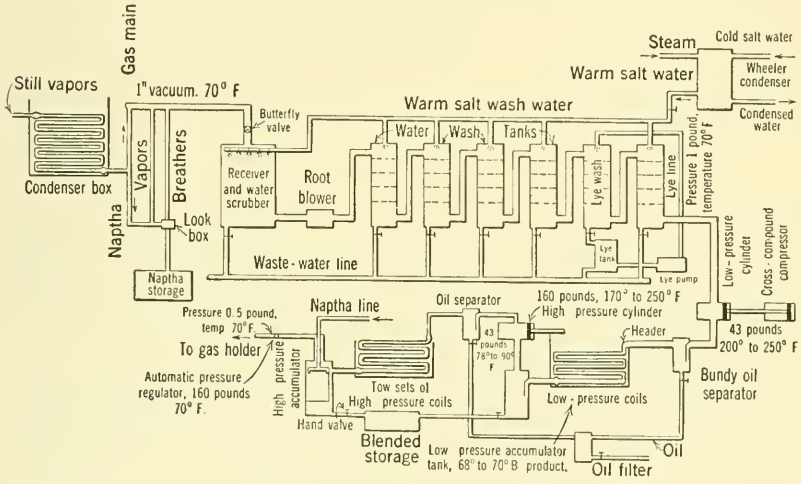



Fig. 375.—Gasoline Manufacture.

into the heat exchanger, then to a pump which forces it through some water cooled atmospheric coils, and finally to the top of the tower from where it began, thus completing the cycle. In the case of natural gas the conditions are the same, practically, except for the characteristics of the gas.

In the case of the absorption process in order to obtain the maximum efficiency of extraction it is necessary to pump a large amount of oil through the tower. This results in a low saturation of the mineral seal oil, a large capacity heat exchanger, a large still and a large pump. Even then it is sometimes impossible to extract the entire gasoline content of the gas in summer time. This is particularly true of plants treating so-called dry or natural gas containing from one-tenth to five-tenths of a gallon per 100 cu. ft. of gas.

In these plants it has been found necessary to use refrigeration to cool the oil down to the temperature at which the vapor tension of the gasoline content of the gas is sufficiently overcome to cause the condensation and absorption by the mineral seal oil absorbent. Therefore, this calls for the introduction of machinery. Also, in the operation of the absorption plants the lighter and more volatile gasoline fractions are not condensed as they are driven off from the still, but pass through with what condensate does form in the absorption accumulator tank, from which point they are withdrawn by a compressor that compresses them up to 80 to 100 lb. After leaving this compressor they pass through a series of water-sprayed condenser coils where they are condensed and separated from the entrained methane and ethane.

It has been found that a gasoline produced in an absorption plant does not need blending, the vapor pressure being low enough so as to allow it to be shipped just as it comes from the still. At some plants gas compressors are used to treat the uncondensed still vapors which come through the cooling coils and make rather a "wild" gasoline of the same characteristics as that produced by other compressor plants. The product is often blended with still run gasoline. Other plants treat the uncondensed still vapors in a secondary absorption tower using mineral seal oil or naphtha as the absorbent. In case mineral seal oil is used, it is sent to the still. If naphtha is used the product is sent to market as it comes from the absorption tower.

The refrigerating machine therefore has a place in gasoline production, although it is not used universally, nor is it of equal advantage in every case. Figures 374 and 375 show the methods used in this kind of work.

**Brewery Refrigeration.**—The brewery was the first important application of refrigeration in the United States, and for a number of years it was the largest user of mechanical refrigeration, having replaced the use of ice and ice and salt in this industry. Since 1919 the industry has suffered very badly in the United States, and many of the 1225 plants requiring 172,800 tons of refrigeration have been modified into other industries or have been closed.

Refrigeration in lager beer brewing is required for the following purposes:

(*a*) To cool the liquid resulting from the malt and hops (wort) from about 70 to 40 deg. F.

(*b*) To cool the wort, at one period or another, from 40 to 32 degrees.

(*c*) To neutralize the effect of fermentation.

(*d*) To cool the cellars, yeast rooms, hop stores, etc., kept at 32 to 38 deg. F.

To cool the wort to 40 degrees requires:

$$Q_1 = B \times W \times d \times C \times (t - 40) \text{ B.t.u.,}$$

where

$B$ = the number of barrels;
$W$ = the nominal weight of a barrel (about 360 lb.);
$d$ = the density of the wort (1.05);
$c$ = the specific heat of the wort (0.9).

To cool the wort to 32 degrees requires:

$$Q_2 = B \times 265(40 - 32).$$

To neutralize the heat of fermentation:

The action of fermenting is

$$C_{12}H_{22}O_{11} = 4C_2H_6O + 4CO_2 + 330 \text{ B.t.u. per lb. of maltose.}$$

$$Q_3 = \frac{B \times 0.91 \times (b - b_1)(259 + b) \times 280}{100},$$

where $b$ and $b_1$ = the specific gravity of the wort and ready beer by Balding saccharometer (about 1.055 and 1.015 respectively).

To cool the cellars, etc.: This is a calculation similar to that in cold storage and depends on the heat leakage, etc., through the building material. The temperatures required are nominal and the result is that the suction pressure on the compressor is usually 30 lb. gage or higher.

**Soap Manufacture.**—Refrigeration is used quite extensively in the United States in the manufacture of soap. As the applications vary in the different plants examples will be given as representative of common practice.

In the manufacture of " boraxine," a flaky soap, the hot soap arrives at the refrigerating apparatus at about 212 deg. F. The usual chilling device is the roll, revolving at approximately 10 r.p.m. similar to the lard and compound rolls (Fig. 300). When brine is used for cooling it enters at 10 deg. and leaves at about 18 deg. F., and if direct expansion is used the evaporating temperature is about 15 deg. F. By the action of the roll the flaky boraxine is chilled to about 32 degrees and the solid soap is scraped off the surface of the roll by means of a knife. The refrigerating load is similar, except for the physical constants, to that given in Chapter XIV.

In making certain toilet soaps it was the practice at one time to fill large drums containing several hundred pounds of the liquid soap and this was permitted to air cool until the entire mass was solid, but this method was found to be both slow and awkward. The modern bar

method is shown in Fig. 376. This is an endless chain of molds, and means are provided by which the molds are made to pass through the long refrigerating box, the speed of the chain and the length of the box being such as will permit a complete hardening of the soap before a complete circuit can be made. The refrigeration is that due to the heat leakage and the cooling and solidifying of the bars. About 15 lb. gage is usually carried in the evaporating coils in the refrigerated box.

**The Manufacture of Rubber.**—In the manufacture of rubber, refrigeration is used to assist in the removal of the heat generated in the mill rolls caused by the mixture of rubber with its compounds on the surface of the rolls. The surface speed ratio of the two rolls through

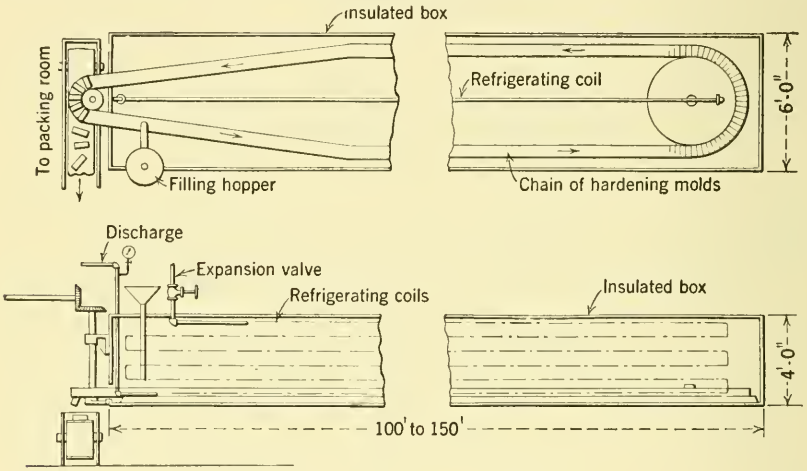



Fig. 376.—Chilling of Bar Soap.

which the rubber passes is 1 : $1\frac{1}{2}$; the variation in speed being required to effect the kneading or milling which is desired. The line shaft is attached to the faster roll, and the rubber sticks to the slower one. The operator cuts the rubber off the roll and returns it to the hopper for more kneading. For this purpose the rolls usually are

16 in. diameter by 42-in. face with shell thickness of 4 in.
20 in. diameter by 60-in. face with shell thickness of 5 in.
24 in. diameter by 84-in. face with shell thickness of 6 in.

Nearly all the power delivered to the rolls tends to heat the rubber. To prevent burning, the roll surfaces are limited to not more than 160 deg. F., and this necessitates a water temperature of 58 degrees or lower entering and 64 degrees or lower leaving the rolls. Roughly, the heat which must

be absorbed by the water is 0.9 of the heat equivalent of the power delivered to the shaft or, roughly, 0.9 hp. per 1 in. of roll face.

**Problem.**—Tests show that a motor driving a roll requires 30 kw. What refrigeration is required?

$$\text{Refrigeration} = \frac{30 \times 0.85 \times 0.9 \times 1.34 \times 42.4}{200} = 6.52 \text{ tons.}$$

Formerly refrigeration was used for making it possible to cut block rubber into thin sheets, after the block had been frozen. This method is obsolete at the present time.

**Gelatine Emulsion.**—Large chill rooms are maintained at from 35 to 40 deg. F. for the purpose of chilling emulsion. This converts it from a warm gelatine solution to a solid jelly, and it is best kept in a solid condition for various lengths of time. For photographic work the emulsion is melted at from 90 to 100 deg. F. and coated on a paper or a film base after which it is submitted to a sudden chill. Refrigeration in industry of this nature consists in humidity control and a very careful temperature regulation.

**Glue.**—Both glue and gelatine solutions are cooled by means of refrigeration to 50 or 40 deg. F. The glue liquid, containing approximately 25 per cent dry glue, is sometimes chilled in small pans containing 72 lb. ( 6 in. by 9 in. by 36 in.), or the glue solution may be fed on to belts (Fig. 324), so arranged that the proper temperature will be attained before the solution leaves the chill chamber.

**Refrigeration in Gas Warfare.**—Refrigeration in gas warfare[2] or in general in the preparation of poison and some noxious gases may involve:

(*a*) The cooling of the gas.

(*b*) The liquefying of the gas.

(*c*) The cooling of the liquid.

(*d*) The cooling of the metal containers required for the storage of the condensed gas to a temperature below 32 deg. F.

In the case of phosgene gas, formed by bringing equal volumes of CO and $Cl_2$ together in the presence of a catalyzing agent, the gas was cooled in a double pipe water cooler, then cooled and condensed by passing through helical lead coils submerged in brine where liquefaction occurred at 40 deg. F. Mustard oil is stated to require 2 tons of refrigeration per one ton of oil per day of 24 hours.

**The Haber Process.**—Ammonia is now being made by the fixation of atmospheric nitrogen by the Haber process at 100 to 200 atm. pressure, using a mixture of water and producer gas; the Casale, at 600 to

[2] A. M. Heritage, Journal Amer. Soc. Refrigerating Engineers, May, 1919.

700 atm., using a mixture of electrolytic hydrogen and nitrogen obtained by burning the oxygen from the air with excess hydrogen; and the Claude, at 900 atm. pressure exerted on a mixture nitrogen from the liquid air process and hydrogen from coke oven gas. One plant, at Syracuse, N. Y., is working on a modified Haber process similar to that used by the United States Government Nitrate Plant No. 1 during the World War, and has a capacity of 10 tons per day (1925). According to Frederick Pope,[3] the cost of synthetic ammonia can be reduced to 4 cents per lb. in large plants. The following method has been used in the calculation of the required refrigerating effect in the Haber process.

**Problem.**—90,000 cu. ft. per hour of permanent gases ($3H_2 - N_2$) at 60 deg. F. and 1 atm., under 100 atm. and 10 per cent ammonia content leaving the catalyst, is to be cooled in two coolers, one to 10 deg. and the other to − 40 deg. F. The liquefied ammonia will be trapped off after each cooler. The value for $c_p$ for the permanent gases at 100 atm. is to be taken as 1.0. Find the refrigeration required. For gas mixtures

$$p = p_1 + p_2 = 1470 \text{ lb.}$$

$$= p_{\text{saturated ammonia}} + p_{\text{permanent gas}},$$

but

$$\frac{V_1}{V} = \frac{p_1}{p} = 0.1;$$

therefore,

$$1470 = 147 + 1323,$$

and

$$t_{NH_3 \text{ at } 147 \text{ lb.}} = 77.63 \text{ deg. F.}$$

*Liquefaction.*

$$\text{At 10 degrees } \frac{V_1}{V} = \frac{p_1}{p} = \frac{38.51}{1470} = 2.62 \text{ per cent by volume.}$$

$$\text{At } -40 \text{ degrees } \frac{V_1}{V} = \frac{p_1}{p} = \frac{10.41}{1470} = 0.709 \text{ per cent by volume.}$$

*Volume of Permanent Gases.*

At 100 atm. and 77.6 deg. F.:

$$90{,}000 \times \frac{14.7}{1323} \times \frac{460 + 77.6}{460 + 60} = 1035 \text{ cu. ft.}$$

At 100 atm. and 10 deg. F.:

$$90{,}000 \times \frac{14.7}{1431.5} \times \frac{460 + 10}{460 + 60} = 833 \text{ cu. ft.}$$

[3] Frederick Pope, Ammonia Synthesis, Refrigerating Engineering, 1926.

At 100 atm. and −40 deg. F.:

$$90{,}000 \times \frac{14.7}{1460} \times \frac{460 - 40}{460 + 60} = 732 \text{ cu. ft.}$$

*Weight of Ammonia Present* (lb.)

At 77.6 deg. F. Total volume ÷ specific volume of $NH_3 = \frac{1033}{2.034} = 508$ lb.

At 10 deg. F. $= \frac{833}{7.304} = 114$ lb.

At −40 deg. F. $= \frac{732}{24.86} = 29.4$ lb.

*Quality of Vapor Leaving Refrigerating Coils.*

At 10 deg. F. 114.0 ÷ 508 = 22.45 per cent (by weight).

At −40 deg. F. 29.4 ÷ 114.0 = 26.3 per cent (by weight).

*Weight of Permanent Gases.*

Taking the specific volume, $3H_2 - N_2 = 44.24$ at NTP,

$$\frac{90{,}000}{44.24} = 2034 \text{ lb.}$$

*Ammonia Content* (by weight).

At 77.6 deg. F. $\frac{508}{2034 + 508} = 20.1$ per cent.

At 10 deg. F. $\frac{114}{2034 + 114} = 5.33$ per cent.

At −40 deg. F. $\frac{29.4}{2034 + 29.4} = 1.43$ per cent.

*Refrigeration.*—As the cooling takes place at constant pressure, the heat absorbed by the evaporating coils is equal to the difference in the value of $i$.

$$Q = i_{1_{\text{gas}}} + i''_{1_{\text{Ammonia}}} - (i_{2_{\text{gas}}} + i'_{2_{NH_3}} + x_2 r_{2_{NH_3}})$$

$$= 2034 \times 1.0 \times [77.6 - 10] + 508\ [(630.3 - 53.8) - (0.225 \times 561.1)]$$

$$= 366{,}300 \text{ B.t.u.} = 30.5 \text{ tons.}$$

Likewise, the refrigeration in the second stage cooler at a temperature of −40 deg. F. becomes:

$$Q = (2034 \times 1.0 \times 50) + (614.9 - 0 - 0.263 \times 597.6)$$

$$= 167{,}600 \text{ B.t.u.} = 13.95 \text{ tons.}$$

In the first stage the evaporating pressure, using ammonia as the refrigerant, would be at 15 lb. gage, whereas it would be at 8 lb. absolute in the second stage. The percentage of ammonia in the gas phase as found experimentally is not exactly as given in the calculations, but this method is given in full as an example of what can be done.

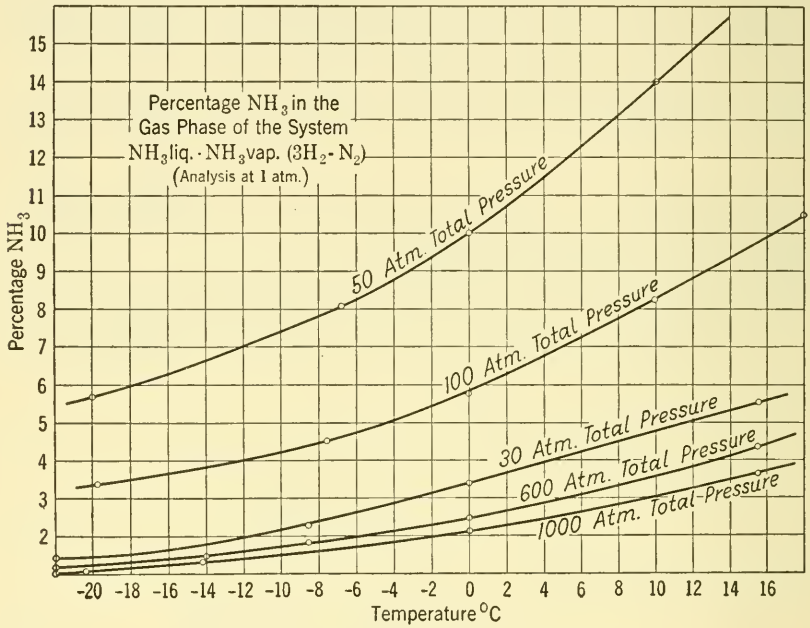



Fig. 377.—Synthetic Manufacture of Ammonia.

Figure 377 gives more accurately the percentage of ammonia in the gas phase of the system $3H_2 - N_2$. This was developed by the fixed nitrogen section of the Department of Commerce. The calculation is given more as a method than as a means of arriving at an accurate answer.

## CHAPTER XIX

## SAFETY DEVICES AND FIRE PROTECTION

Because of the nature of the refrigerants used, the operating cycle, and the pressures encountered, a number of special regulations have been devised for refrigerating plants. These regulations consist in means for prevention of excess pressures in the discharge lines from the compressor, the condenser or the liquid receiver, or (in special cases, as in the case of fire) of discharging the whole of the refrigerant into the atmosphere or a suitable mixing chamber.

**The Safety Valve.**—The safety valve for ammonia is simply a pop valve ground to its seat, whereas the carbonic valve has a thin metal disc usually in addition to the pop valve. The valve is best piped to discharge into the low-pressure side of the system and should be placed on the shell type brine cooler, the shell type condenser, and the liquid receiver where the pressure can become excessively high on the occasion of the temperature rising with both the inlet and the outlet valves closed. If a pop valve blows, it must be reground in order to be put back into service. When the discharge is into a mixer (Fig. 381), or into the atmosphere, the refrigerant is lost.

When the compressor is *electrically driven* it is usual to have special devices to stop the compressor should the pressure into the discharge line rise to an excess: for example, as high as 250 lb. in the case of ammonia. In Fig. 378 the gas pressure exerted on the diaphragm compresses a spring and moves a plunger, thereby releasing a spring operated trip which is connected to a knife switch. Various valves of similar design are manufactured.

Should a cylinder head break, or similar trouble develop, a non-return valve comparable to those used in steam boiler headers would be very valuable, and such valves are installed in large plants in order to prevent the loss of the entire charge and the consequent vitiation of the compressor room and the vicinity. A hand-operated valve is shown in Fig. 379, and this can be modified to be operated by means of a push button. In either case a remote operation of the stop valve is possible.

**Helmets.**—Every plant above the fractional tonnage size should be provided with suitable gas masks or helmets in good working order.

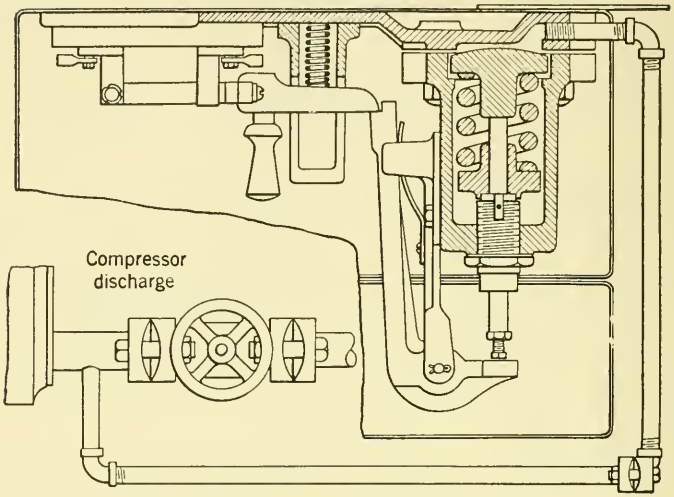



FIG. 378.—Pressure Actuated Cut Out.

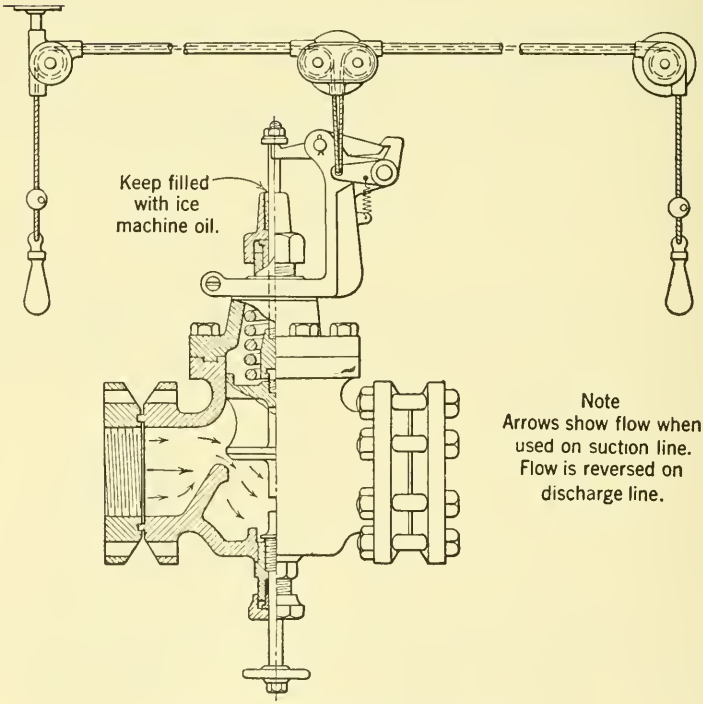



FIG. 379.—Remote Control Stop Valve.

They should be placed *outside* the compressor room, but within easy access at all times.

**Open Flames.**—With ammonia, ethyl and methyl chloride and the hydro-carbon refrigerants no flames, arc lights, gas jets or any apparatus employing a flame except internal combustion engines with internal ignition should be permitted. It will be seen in Chapter VII that the refrigerants listed are explosive with certain limited mixtures and, as a matter of fact, several bad explosions have been laid to the door of ammonia.

**Inspection of Refrigerating Plants.**—The City of Chicago inspection rules for refrigerating plants requires the inspector to note particularly the following 10 points:

1. Valves and fittings entering the construction of machinery, apparatus, pipes and the connections therewith, and wherein ammonia is used under pressure, shall be subjected to an air pressure test of 300 lb., and 500 lb., of hydrostatic pressure.
2. Hangers and braces shall be of flat angle or round wrought iron or steel.
3. No perforated material will be accepted.
4. Supports and brackets may be of cast iron.
5. The discharge pipe between compressor or generator and condenser sections shall be extra heavy material and be provided with two stop valves.
6. All liquid pipes must be of extra heavy material.
7. High and low-pressure check valves, or equivalent, must be located as close to compressor as convenient, and provided with a by-pass. Check valves in absorption plant shall be located between rectifier and condenser and in the discharge line close to the aqua ammonia pump.
8. Safety pop valve must be located between compressor or generator and first stop valve and arranged for seal and set at 250 lb. for ammonia compressors and 1500 lb. for $CO_2$ compressors.
9. Size of safety valve for 250 lb. pressure:

Capacity not exceeding 10 tons, $\frac{1}{2}$ in.
Capacity over 10 tons and not exceeding 25 tons, $\frac{3}{4}$ in.
Capacity over 25 tons and not exceeding 40 tons, 1 in.
Capacity over 40 tons and not exceeding 60 tons, $1\frac{1}{4}$ in.
Capacity over 60 tons and not exceeding 100 tons, $1\frac{1}{2}$ in.
Capacity over 100 tons and not exceeding 140 tons, 2 in.
Capacity over 140 tons and not exceeding 190 tons, $2\frac{1}{2}$ in.
Capacity over 190 tons and not exceeding 300 tons, 3 in.
Tonnage based on 7,500 cu. in. displacement per ton for ammonia compressor.

10. Separators, liquid receivers and receptacles on the high pressure side of an ammonia machine, shall be tested to a hydrostatic pressure of 500 lb. and 300 lb. of air. Hydrostatic test to be stamped on each vessel with name of maker, shell brine coolers, purifiers, absorbers, carrying a maximum working pressure below 250 lb., shall be equipped with a safety valve not less than $\frac{1}{2}$ in. diameter, and set for a pressure arrived at by computing the material and construction of each apparatus according to the rules applied thereon. Liquid receivers must be equipped with a $\frac{1}{2}$ in.

pop safety valve, and a stop valve between safety valve and receiver. Arrange stop valve so it can be sealed when open. Outlet of safety valve to be piped to atomsphere outside of building.

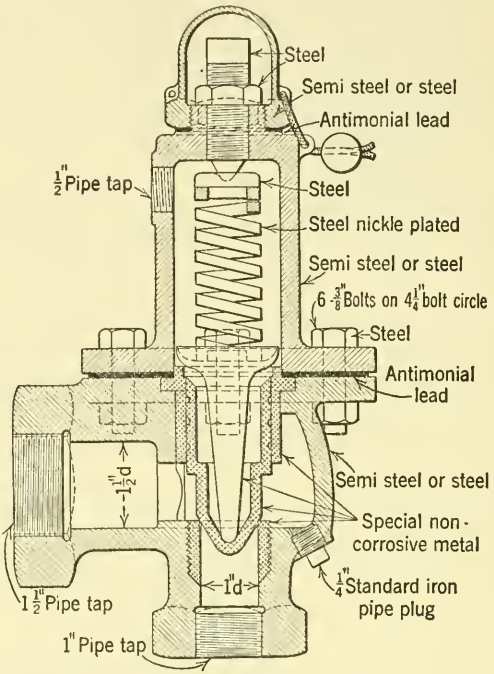



Fig. 380.—Pop Safety Relief Valve.

**The Mechanical Refrigerating Safety Code.**—A safety code for refrigerating machinery has been devised by the American Society of Mechanical Engineers in conjunction with the American Society of Refrigerating Engineers (1925),[1] under the direction of the American Engineers Standards Committee, which is as follows:

### MECHANICAL REFRIGERATION SAFETY CODE [2]

#### SECTION 10

##### Scope

This code shall apply to every refrigerating equipment employing fluids which are vaporized and liquefied or compressed in their refrigerating cycle and is intended

[1] See also the A.S.M.E. Boiler Code, Part I, Section 6, for rules for the construction of unfired pressure vessels.

[2] Formulated under direction of American Engineering Standards Committee and sponsorship of The American Society of Refrigerating Engineers.

to provide for the safe installation, operation and inspection of refrigerating systems, as well as the storage of refrigerants.

### Interpretations and Exceptions

The purpose of this code is to provide reasonable safety for life, limb, health and property. It shall be liberally construed to secure these results by the enforcing officers or body, who shall have authority in cases of practical difficulty or unnecessary hardship to grant exceptions from the literal requirements of this code, as long as equivalent protection is thereby secured. When the safeguarding of specific types of machines is covered by other approved codes, these codes shall be given preference. When specific devices or methods are mentioned in this code, other devices or methods which will secure equally good results may be used, subject to the approval of the administrative authority.

## SECTION 20

### Definitions

(*a*) *Refrigerating Equipment.*—A "refrigerating equipment" or "refrigerating system" is an apparatus, including its source of energy in so far as it affects its safety of operation, for abstracting heat from a body at one temperature and transferring it to another at a higher temperature.

(*b*) *Capacity.*—The "capacity" of a refrigerating equipment is its rating in tons of refrigeration per twenty-four (24) hours, determined as follows:

*a'*. A standard ton of refrigeration is two hundred and eighty-eight thousand (288,000) B.t.u.

*b'*. The standard commercial ton of refrigeration is at the rate of two hundred (200) B.t.u. per minute.

*c'*. The standard rating of a refrigerating machine, using liquefiable vapor, is the number of standard commercial tons of refrigeration it performs under the following adopted refrigerant pressures:

(1) The inlet pressure is that which corresponds to a saturation temperature of five degrees (5°) F. (−15° C.).

(2) The outlet pressure is that which corresponds to a saturation temperature of eighty-six degrees (86°) F. (30° C.).

These pressures are measured outside and within ten (10) feet of the pressure imposing element, measured along the inlet and outlet pipes, respectively.

(*c*) *Working Pressure* shall be understood to mean the maximum allowable working pressure.

(*d*) *Pressure Imposing Element* is that part of a refrigerating equipment which draws the refrigerant from the low-pressure side and discharges it into the high-pressure side of the system, such as a compressor or absorber and generator.

(*e*) *Machinery Room* is one in which is located any pressure-imposing element, condenser, receiver or shell-type apparatus. Rooms which contain only pipes for conveying the refrigerant are not considered machinery rooms.

(*f*) *Pressure Limiting Device* is a device acting directly responsive to pressure to stop the action of the pressure-imposing element.

(*g*) *Pressure Relief Devices*—

*a'*. *Pressure Relief Valve* is a device that will automatically relieve the excess pressure at a predetermined maximum difference of pressure.

*b'. Automatic By-Pass* is a device that will automatically relieve within narrow limits of a predetermined pressure the excessive pressure, irrespective of the outlet pressure.

*c'. Fusible Plug* is a device which will, by the fusing or softening of a suitable material, relieve the pressure when the device has reached a predetermined temperature.

*d'. Rupture Member* is a device that will rupture at a predetermined difference of pressure.

(*h*) *Vessel* is any part of a system containing refrigerant.

(*i*) *Liquid Receiver* is a liquid refrigerant reservoir permanently connected by inlet and outlet pipes to a refrigerating system.

(*j*) *Container* is a cylinder or drum for the shipment of a refrigerant.

(*k*) *Suction and Discharge Stop.*—The terms "main suction stop" and "main disharge stop" designate the manually operated shut-off valves located in the suction and discharge lines, respectively, adjacent to the pressure imposing element.

(*l*) *Service Valve* is a valve located in the refrigerant circuit which may be operated only by a specially constructed service key.

## SECTION 30

### General Requirements

(*a*) Pressure relief and pressure limiting devices shall be made of material suitable for the refrigerant employed and, where practicable, their working parts shall be non-corrodible and shall be set, marked and sealed by the manufacturer.

(*b*) Automatic by-passes shall be of such design as to operate within five per cent (5%) of the pressure for which stamped, irrespective of the outlet pressure.

(*c*) If rupture member devices are used, they shall be rupturable by a direct pressure. The rupturing pressure shall be legibly marked on the rupturing member or on proper attachment thereof.

(*d*) There shall be no valve between any pressure relief devices and the receivers, condensers, evaporators, or other vessels which they protect or between automatic by-pass devices and the main suction line. There shall be also no valve between any rupture member or fusible plug or other pressure limiting means and the part of the equipment protected thereby, except that service valves may be permitted on Class C Equipment.

(*e*) All piping, and liquid receivers containing the refrigerant, shall be so installed as to be least liable to damage in case of accident.

(*f*) Electrical equipment shall be installed in accordance with the National Electrical Code.

(*g*) In Class A and Class B equipment there shall be a service switch, or remote control switch, controlling all the electrically driven refrigerating machinery, one control of which, properly labeled, shall be located outside of the machinery room, where it can be quickly and safely reached and operated in case of an emergency.

(*h*) Switchboards and high tension cables shall be located where they will be least liable to damage if an accident occurs to the refrigerating equipment.

(*i*) All working parts of refrigerating machines, prime movers and drive belts shall be guarded in accordance with the provisions of the National Safety Code for Mechanical Transmission of Power.

## SECTION 30

### Operating Precautions

(*a*) Containers having refrigerant therein shall be stored in a cool, well ventilated place, as remote as possible from danger by heat or fire.

(*b*) Refrigerant not contained in the refrigerating system shall be stored only in containers conforming with the regulations of the Interstate Commerce Commission for the transportation of such refrigerant.

(*c*) When refrigerant is withdrawn from the system, it shall be discharged to the outside atmosphere or suitable absorbent or stored in containers conforming with the regulations of the Interstate Commerce Commission for the transportation of such refrigerant. Care shall be taken to prevent escape of refrigerant into building.

(*d*) In withdrawing refrigerant from the equipment into containers, they shall be placed on a scale and shall not be filled to more than allowed capacity as prescribed by Interstate Commerce Commission regulations.

(*e*) Containers shall not be connected to the system except during period of charging.

(*f*) Repairs to refrigerant lines or apparatus shall not be made while they are under pressure. Tightening of bolts or flanged joints shall be done at a reduced pressure.

(*g*) In testing systems with air pressure care shall be taken to prevent the temperature at any point rising above one hundred and thirty degrees (130°) F. (hand warm).

## Part I—Ammonia Systems

## SECTION 100

### General Requirements for Ammonia Systems

(See also General Requirements, Section 30.)

(*a*) Every part of the high pressure side of the system shall be designed for a maximum allowable working pressure of at least two hundred (200) pounds per square inch gauge pressure and every part of the low-pressure side shall be designed for a maximum allowable working pressure of at least one hundred and fifty (150 pounds) per square inch gauge pressure. A factor of safety of five (5) shall be used in all designs, except for gauges and control mechanism.

(*b*) Every vessel in the system, except connecting piping, shall be marked with the maximum allowable working pressure and shall be tested hydrostatically to one and one-half (1½) times this pressure marking, such tests to be conducted in accordance with the "Unfired Pressure Vessel Code."

## SECTION 110

### Class A Equipment

A "Class A Equipment" is a refrigerating system of thirty (30) tons capacity or over, or containing one thousand (1000) pounds of ammonia or over.

*Rule* 1100.—*Pressure Imposing Element.*

(*a*) Every equipment shall be provided with one or more automatic by-passes of proper size, connected between each main discharge stop valve and the pressure

imposing element, to relieve excessive pressures into the low pressure side, on either side of the main suction stop valve; or, as an alternative, shall be provided with one or more pressure relief valves of proper size discharging, during the period of excess pressure, to the outside atmosphere (see Rule 1103 (*a*)), or to the low-pressure side when a pressure relief valve is installed thereon discharging to the outside atmosphere. These pressure relief devices shall be so designed and installed as to prevent pressures in the high-pressure side exceeding the maximum allowable working pressure. The proper (nominal pipe) sizes of such devices shall be as follows:

| | |
|---|---|
| For equipment of from | 30 to 60 tons capacity, use at least one $\frac{3}{4}$ in. device. |
| | 60 to 100 tons capacity, use at least one 1 in. device. |
| | 100 to 175 tons capacity, use at least one $1\frac{1}{4}$ in. device. |
| | 175 to 250 tons capacity, use at least one $1\frac{1}{2}$ in. device. |
| | 250 to 450 tons capacity, use at least one 2 in. device. |
| | 450 to 900 tons capacity, use at least two 2 in. devices. |

(*b*) Every equipment shall have an automatic pressure-limiting device, acting directly responsive to pressure, to stop the action of the pressure-imposing element at a pressure not higher than ninety per cent (90%) of the pressure stamped on the pressure relief valve.

(*c*) A cushioned or other satisfactory check valve shall be placed in the discharge line between the pressure-imposing element and the condenser.

*Rule* 1101.—*Liquid Receiver, Condenser, Evaporator.*

(*a*) Each liquid receiver, shell type condenser and shell type evaporator, which can be isolated, shall be equipped with a one-half ($\frac{1}{2}$) inch automatic by-pass set to discharge at a pressure not higher than the maximum allowable working pressure, and connected at the highest point to discharge to that part of the low-pressure side of the equipment protected by an emergency relief device. As an alternative, a pressure relief valve discharging to the atmosphere may be used. (See Rule 1103 (*a*).)

*Rule* 1102.—*Fire Emergency Devices.*

(*a*) The low-pressure side of every equipment shall be provided with a hand-operated relief valve for discharging the ammonia in case of fire, either to the atmosphere, to a suitable body of water or through a mixer.

(*b*) The hand-operated relief valve shall be located on the outside of the wall of the engine room, or shall be capable of operation therefrom. The valve or operating mechanism shall have a protecting cover.

(*c*) The minimum (standard pipe) sizes of the hand relief valves shall be as follows:

| | |
|---|---|
| Where the charge is | 1,000 to 1,800 lb., use one $\frac{3}{4}$ in. valve. |
| | 1,800 to 3,000 lb., use one 1 in. valve. |
| | 3,000 to 5,250 lb., use one $1\frac{1}{4}$ in. valve. |
| | 5,250 to 7,500 lb., use one $1\frac{1}{2}$ in. valve. |
| | 7,500 to 13,500 lb., use one 2 in. valve. |
| | 13,500 to 27,000 lb., use one 3 in. valve. |

*Rule* 1103.—*Refrigerant Discharge.*

(*a*) When discharged into the atmosphere, the ammonia shall be conducted by continuous piping to an outlet turned upward and equipped with a suitable diffuser, designed to mix the ammonia with air. The piping from the hand relief valve to

point of final discharge, including the diffuser, shall be such as not to impose a resistance head of more than ten (10) pounds per square inch on the discharge side of the valve. Such piping shall be used only for ammonia discharge and be so located that the gases discharged are not liable to become ignited. Outlets from diffusers shall be above the roof of any building within fifty (50) feet.

(*b*) Outlets from atmospheric pressure relieve devices shall connect to the discharge pipe specified in Paragraph (*a*) of this rule, or a similar pipe.

(*c*) In localities where the discharge of a refrigerant in case of fire comes under the direction of the fire department, the hand relief valve shall be located in a locked box, which can be opened only by members of the fire department by means of a fire department key. The door on this box shall be on the public thoroughfare side of the building wall. If this is impracticable, the door shall be in a vestibule having glass panel doors, which provide easy access from the street. The door of the box shall be at a height of not more than five (5) feet nor less than two (2) feet above the street or vestibule floor level. In either case it shall be in a place as remote as possible from any exhaust vent or outlet from the machinery room. At the discretion of the fire department, such box may be equipped with an auxiliary door, also locked, to be opened only by such persons as may be duly authorized by the fire department to make repairs.

(*d*) The protecting box shall be plainly labeled "For Fire Department Use Only." The emergency valve shall be labeled "Ammonia." In case the ammonia is to be discharged into the sewer, as provided in Paragraph (*c*), there shall be also within the box a sign reading "Do not open valve until water is flowing into the mixer."

(*e*) When the local fire department requires that a refrigerant be discharged through a mixer into the sewer, there shall be provided a check valve and a standard fire department connection through which the fire department supplies the necessary water under the proper pressure. A stop valve shall be installed inside the building in the refrigerant line leading to the mixer hand relief valve for use in repairing same. This valve shall be sealed open, except for and during repairs.

(*f*) The fire department shall make at least an annual inspection of valves, emergency means and mixer.

*Rule* 1104.—*Helmets or Masks.*

(*a*) At least two (2) helmets or masks of approved make, tested in accordance with the requirements of the United States Bureau of Mines for ammonia, shall be kept in operative condition and available for the immediate use of responsible attendants, who shall be trained in their use.

(*b*) Helmets or masks shall be kept in a place easily accessible from outisde the machinery room.

*Rule* 1105.—*Open Flames.*

(*a*) In machinery rooms, there shall be no fire, arc light, hot surface ignitor or flame open to the atmosphere of the room; any device intended to adequarely isolate the fire, arc light, hot surface ignitor or flame shall have been approved by a competent laboratory and the administrative authority. Internal combustion engines with hot surface ignition may be started in the usual manner.

*Rule* 1106.—*Electrical Equipment.*

(*a*) All current carrying devices, including bus bars, operating at above six hundred (600) volts shall have all live parts in tight enclosures, or be provided with suitable insulating covering.

(*b*) Extensive electrical equipment other than that essential to the ice making or refrigerating plant shall not be permitted in the machinery room.

(*c*) Transformers with primary or secondary connections above six hundred (600) volts shall be in locked vaults which, if adjoining the machinery room, shall be separated therefrom, preferably by unpierced walls, ceilings and floors of six (6) inches of reinforced concrete or its equivalent. All conduit ends and wall openings for conduits entering the vault shall be sealed. (See also the National Electrical Code.)

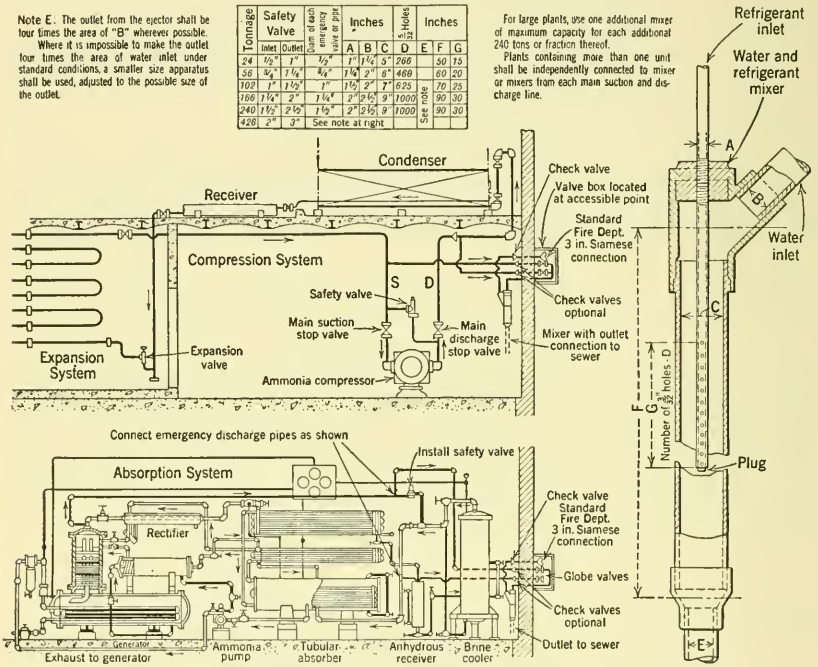

Note E. The outlet from the ejector shall be four times the area of "B" wherever possible. Where it is impossible to make the outlet four times the area of water inlet under standard conditions, a smaller size apparatus shall be used, adjusted to the possible size of the outlet.
Tonnage
Safety Valve
Inlet
Outlet
Diam. of each discharge pipe
Inches
A
B
C
D
No. of Holes
Inches
E
F
G
24 1/2" 1" 1/2" 1" 1 1/4" 5" 266 30 15
36 3/4" 1 1/4" 3/4" 1 1/4" 2" 6" 466 60 20
100 1" 1 1/2" 1" 1 1/2" 2" 7" 625 70 25
166 1 1/4" 2" 1 1/4" 2" 9 1/2" 9" 1000 90 30
240 1 1/2" 2 1/2" 1 1/2" 2" 2 1/2" 9 1000 90 30
426 2" 3" See note at right
See note
For large plants, use one additional mixer of maximum capacity for each additional 240 tons or fraction thereof.
Plants containing more than one unit shall be independently connected to mixer or mixers from each main suction and discharge line.
Refrigerant inlet
Water and refrigerant mixer
A
B
C
Water inlet
Condenser
Receiver
Check valve
Valve box located at accessible point
Standard Fire Dept. 3 in. Siamese connection
Compression System
S
D
Safety valve
Main suction stop valve
Main discharge stop valve
Check valves optional
Mixer with outlet connection to sewer
Expansion System
Expansion valve
Ammonia compressor
Number of 3/32 holes: D
F
G
Plug
Connect emergency discharge pipes as shown
Install safety valve
Absorption System
Rectifier
Check valve
Standard Fire Dept. 3 in. Siamese connection
Globe valves
Check valves optional
Outlet to sewer
E
Generator
Exhaust to generator
Ammonia pump
Tubular absorber
Anhydrous receiver
Brine cooler

FIG. 381.—Ammonia Mixer Valve.

*Rule* 1107.—*Gauges, Gauge Glasses.*

(*a*) Every equipment shall be provided with pressure gauges; one for indicating pressure in condenser, or high-pressure side, and one for indicating pressure in evaporator, or low-pressure side.

(*b*) Refrigerant level gauge glasses, if used, shall have automatic closing shut-off valves. Such glasses shall be protected by enclosing them in metal casings having longitudinal slots on two opposite sides. The walls of the casing shall be at least one-sixteenth ($\frac{1}{16}$) inch thick and so supported that impacts on the casing are not liable to be transmitted to the glass.

*Rule* 1108.—*Exits, Ventilation.*

(*a*) Each machinery room shall have a direct exit to the open air, or to a room or hall from which gases from machinery room can be excluded by close fitting self-closing door.

(*b*) Vertical and horizontal openings that permit the passage of gases to other parts of the building, which are not a part of the refrigerating plant, shall be sealed or provided with close fitting self-closing door.

(*c*) Each machinery room shall be provided with means for adequate ventilation to the open air, either direct or by means of a suitable duct or ducts. Where a mechanical system of ventilation is employed, the control of such system shall be easily accessible and located outside of the machinery room.

*Rule* 1109.—*Charging.*

(*a*) Every equipment shall be charged through a connection located on the low pressure side.

## SECTION 120

### Class B Equipment

A "Class B Equipment" is a refrigerating system of less than thirty (30) tons capacity, or containing less than one thousand (1000) pounds and over fifteen (15) pounds of ammonia.

*Rule* 1200.—*Pressure Imposing Element.*

(*a*) Every equipment, except that which cannot be subjected to a pressure in excess of the maximum allowable working pressure by the action of the pressure-imposing element, shall be provided with a one-half ($\frac{1}{2}$) inch automatic by-pass connected between each main discharge stop valve and the pressure-imposing element to relieve excessive pressure into the low-pressure side on either side of the main suction stop valve; or as an alternative through a one-half ($\frac{1}{2}$) inch (standard pipe) pressure relief valve discharging as specified in Rule 1203 (*a*), or to the low-pressure side. These pressure relief devices shall be so designed and installed as to prevent pressures in the high-pressure side exceeding the maximum allowable working pressure.

(*b*) Every equipment, except that which cannot be subjected to a pressure in excess of the maximum allowable working pressure by the action of the pressure-imposing element, shall have an automatic pressure-limiting device, acting directly responsive to pressure, to stop the action of the pressure-imposing element at a pressure not higher than ninety per cent (90%) of the pressure stamped on the relief valve.

*Rule* 1201.—*Liquid Receiver, Condenser, Evaporator.*

(*a*) Each liquid receiver, shell type condenser and shell type evaporator, which can be isolated, shall be equipped with a one-half ($\frac{1}{2}$) inch automatic by-pass, set to discharge at a pressure not higher than the maximum allowable working pressure, and connected at the highest point, to discharge to that part of the low-pressure side of the equipment protected by an emergency relief device. As an alternative, a pressure relief valve discharging to the atmosphere may be used. (See Rule 1203 (*a*).)

*Rule* 1202.—*Fire Emergency Devices.*

(*a*) Every equipment shall be provided with a one-half ($\frac{1}{2}$) inch pressure relief valve connected to the low-pressure side of the equipment, set to relieve excessive pressure in a manner specified in Rule 1203 (*a*), and shall discharge at a pressure not higher than the maximum allowable working pressure.

(*b*) Class B Equipment containing less than one hundred (100) lb. of ammonia and so designed or provided with devices as to prevent the existence of pressures in any part of the equipment in excess of the maximum allowable working pressure may be protected by one pressure relief valve.

(*c*) Class B Equipment containing less than one hundred (100) lb. of ammonia, in which the design of the pressure-imposing element is such that the generation of pressure in excess of the maximum allowable working pressure is impossible, may be protected by fusible plugs on each part of the equipment containing ammonia which can be exposed to heat from an external source, provided pressure generated at the temperature necessary to fuse the plug is less than the maximum working pressure.

*Rule* 1203.—*Refrigerant Discharge.*

(*a*) The ammonia passed by the pressure relief devices shall be discharged either to the atmosphere or to a suitable body of water.

(*b*) When discharged into the atmosphere, the ammonia shall be conducted by continuous piping to an outlet turned upward and equipped with a suitable diffuser, designed to mix the ammonia with air. The piping from the pressure relief device to point of final discharge, including the diffuser, shall be such as not to impose a resistance head of more than ten (10) pounds per square inch on the discharge side of the device. Such piping shall be used only for ammonia discharge and be so located that the gases discharged are not liable to become ignited. Outlets from diffusers shall be above the roof of any building within fifty (50) feet.

(*c*) When the ammonia is discharged into a tank of water, the tank shall not be used for any other purpose. At least one (1) gallon of fresh water for every pound of ammonia contained in the equipment shall automatically be maintained in the tank. Provision shall be made to prevent the water from freezing without the use of salt or other chemicals.

(*d*) The tank shall be substantially constructed with a hinged cover or, if it is of the enclosed type, it shall have both a water inlet and a vent hole at the top. No horizontal dimensions of the tank shall be greater than one-half ($\frac{1}{2}$) the height. The discharge pipe from the pressure relief devices shall be so attached as to discharge the refrigerant at the center of the bottom and that portion of the pipe within the tank shall be of lead with a one-sixteenth ($\frac{1}{16}$) inch vent hole above the water level. The tank shall be securely supported and firmly braced. There shall be no opening in the tank below the water level and the water level shall be at least six (6) inches below the top of tank.

*Rule* 1204.—*Helments or Masks.*

(*a*) At least one (1) helmet or mask of approved make, tested in accordance with the requirements of the United States Bureau of Mines for ammonia, shall be kept in operative condition and available for the immediate use of responsible attendants, who shall be trained in their use.

(*b*) Helmets or masks shall be kept in a place easily accessible from outside the machinery room.

*Rule* 1205.—*Open Flames.*

(*a*) In machinery rooms there shall be no fire, arc light, hot surface ignitor or flame open to the atmosphere of the room; any device intended to adequately isolate the fire, arc light, hot surface ignitor or flame shall have been approved by a competent laboratory and the administrative authority. Internal combustion engines with hot surface ignition may be started in the usual manner.

*Rule* 1206.—*Electrical Equipment.*

(*a*) All current-carrying devices, including bus bars, operating at above six hundred (600) volts shall have all live parts in tight enclosures or be provided with suitable insulating covering.

(*b*) Extensive electrical equipment other than that essential to the ice making or refrigerating plant shall not be permitted in the machinery room.

(*c*) Transformers with primary or secondary connections above six hundred (600) volts shall be in locked vaults which, if adjoining the machinery room, shall be separated therefrom, preferably by unpierced walls, ceilings and floors of six (6) inches of reinforced concrete or its equivalent. All conduit ends and wall openings for conduits entering the vault shall be sealed. (See also the National Electrical Code.)

*Rule* 1207.—*Gauges, Gauge Glasses.*

(*a*) Every equipment containing over one hundred (100) lb. of ammonia shall be provided with pressure gauges; one for indicating the pressure in condenser or high-pressure side and one for indicating the pressure in evaporator or low-pressure side.

(*b*) Refrigerant level gauge glasses, if used, shall have automatic-closing shut-off valves. Such glasses shall be protected by enclosing them in metal casings having longitudinal slots in two opposite sides. The walls of the casing shall be at least one-sixteenth ($\frac{1}{16}$) inch thick and so supported that impacts on the casing are not liable to be transmitted to the glass.

*Rule* 1208.—*Exits, Ventilation.*

(*a*) Each machinery room shall have a direct exit to the open air or to a room or hall from which gases from machinery room can be excluded by a close-fitting self-closing door.

(*b*) Vertical and horizontal openings that permit the passage of gases to other parts of the building, which are not a part of the refrigerating plant, shall be sealed or provided with close-fitting self-closing doors.

(*c*) Each machinery room shall be provided with means for adequate ventilation to the open air, either direct or by means of a suitable duct or ducts. When a mechanical system or ventilation is employed, the system shall be capable of producing a complete change of air at least once in five (5) minutes. The control of such mechanical system shall be easily accessible and located outside of the machinery room.

*Rule* 1209.—*Charging.*

Every equipment shall be charged through a connection located on the low-pressure side.

## SECTION 130

### Class C Equipment

A Class C Equipment is a refrigerating system containing fifteen (15) lb. of ammonia or less.

*Rule* 1300.—*Pressure-Limiting Devices*[3] (*Operating Safety Provisions*).

(*a*) Every Class C Equipment containing more than ten (10) pounds of ammonia and that may be subjected to a pressure in excess of the maximum allowable working

[3] It is recommended that only such safety devices which have been designated as suitable for the purpose by a competent disinterested laboratory be used.

pressure, by the action of the pressure-imposing element, shall be provided with a pressure-limiting device to stop the action of the pressure-imposing element.

(*b*) Such pressure-limiting device shall be so constructed and connected as to act directly responsive to the discharge pressure. There shall be no valve, except necessary service valves, between this device and the pressure-imposing element.

(*c*) Pressure-limiting devices shall be set, marked and sealed to stop the action of the pressure-imposing element at a pressure not in excess of the maximum allowable working pressure, or shall be so designed that they cannot be adjusted to operate at a greater pressure than the pressure stamped thereon.

*Rule* 1302.—*Electrical Equipment.*

(*a*) All electrical equipment shall be installed in accordance with the National Electrical Code.

## Part II—Carbon Dioxide Systems

### SECTION 200

### General Requirements for Carbon Dioxide Systems

(*a*) Every part of the high-pressure side of the system shall be designed for a maximum allowable working pressure of at least twelve hundred (1200) pounds per square inch gauge pressure and every part of the low-pressure side shall be designed for a maximum allowable working pressure of at least six hundred and fifty (650) pounds per square inch gauge pressure. A factor of safety of five (5) shall be used in all designs, except for gauges and control mechanism.

(*b*) Every vessel in the system, except connecting piping, shall be marked with the maximum allowable working pressure and shall be tested hydrostatically to one and one-half ($1\frac{1}{2}$) times this pressure marking, such tests to be conducted in accordance with the "Unfired Pressure Vessel Code."

### SECTION 210

### Class A Equipment

A "Class A Equipment" is a refrigerating system of thirty (30) tons' capacity or over, or containing one thousand (1000) pounds of carbon dioxide or over.

*Rule* 2100.—*Pressure-Imposing Element.*

(*a*) Every equipment shall be provided with one pressure relief valve of proper size connected between each main discharge stop valve and the pressure-imposing element to relieve excessive pressures, during the period of excess pressure, and discharge into the machinery room. These pressure relief valves shall be so designed as to prevent pressures in the high-pressure side exceeding the maximum allowable working pressure. The proper (nominal pipe) sizes of the pressure relief valves shall be as follows:

For equipment of from
- 30 ton to 175 ton, use at least one $\frac{1}{2}$ in. valve.
- 175 ton to 450 ton, use at least one $\frac{3}{4}$ in. valve.
- 450 ton to 900 ton, use at least one 1 in. valve.

(*b*) Every equipment shall have an automatic pressure-limiting device, acting directly responsive to pressure, to stop the action of the pressure-imposing element at a pressure not higher than ninety per cent (90%) of the pressure stamped on the pressure relief valve.

(*c*) A cushioned or other satisfactory check valve shall be placed in the discharge line between the pressure-imposing element and the condenser.

*Rule* 2101.—*Liquid Receiver, Condenser, and Shell Type Evaporator.*

(*a*) Each liquid receiver, battery of double pipe condenser, shell type condenser, and shell type evaporator which can be isolated, shall be equipped with a one-half ($\frac{1}{2}$) inch pressure relief valve set to discharge at a pressure not higher than the maximum allowable working pressure, and connected at the highest point to discharge into the machinery room.

(*b*) The low-pressure side of every equipment, except where the cooling surface is entirely submerged in a liquid, shall be provided with a one-half ($\frac{1}{2}$) inch pressure relief valve designed to prevent pressures in the low-pressure side exceeding the maximum allowable working pressure, and discharging into the machinery room.

(*c*) In place of each relief device (Rule 2101 (*a*) and(*b*)), a rupture member may be employed and its discharge side piped to the outside of the building. The rupture member shall be designed to rupture at the maximum allowable working pressure of the vessel to which it is connected.

(*d*) Piping from rupture members may be joined to a common pipe and thence piped to the outside of the building. No piping from rupture members shall be less than one-half ($\frac{1}{2}$) inch iron pipe size, and for each one hundred (100) feet of run the pipe shall increase one pipe size. The free end of any pipe shall be protected by a suitable gooseneck or screen.

(*e*) The rupture members shall be designed to prevent their rupture parts entering the discharge pipe connections.

(*f*) The rupture orifice for any rupture member shall be not less than one-quarter ($\frac{1}{4}$) inch in diameter and the orifice diameter shall be legibly stamped on the outside of the rupture member body.

(*g*) In place of pressure relief valves mentioned in Rules 2100 (*a*) and 2101 (*a*), (*b*), (*c*), and (*d*) combination devices consisting of a rupture element and a relief valve may be used. These devices shall function at the maximum allowable working pressure. The dimensions of the combination devices shall be the same as specified for relief valves and rupture members, except that no rupture element in a combination device shall have an orifice less than one-half ($\frac{1}{2}$) inch in diameter. The relief element may discharge into the machinery room. This combination device shall be so designed that pressure is exerted against the relief valve only after the rupture element has functioned.

*Rule* 2102.—*Gauges, Gauge Glasses.*

(*a*) Every equipment shall be provided with pressure gauges; one for indicating pressure in condenser, or high-pressure side, and one for indicating pressure in evaporator or low-pressure side.

(*b*) Refrigerant level gauge glasses, if used, shall have automatic closing shut-off valves. Such glasses shall be protected by enclosing them in metal casings having longitudinal slots on two opposite sides. The walls of the casing shall be at least one-sixteenth ($\frac{1}{16}$) inch thick and so supported that impacts on the casing are not liable to be transmitted to the glass.

*Rule* 2103.—*Exits, Ventilation.*

(*a*) Each machinery room shall have a direct exit to the open air, or to a room or hall from which gases from machinery room can be excluded by close fitting door.

(*b*) Each machinery room shall be provided with adequate ventilation, as follows:

(1) Where the machinery room floor level is not more than four (4) feet below grade, natural ventilation, to outside atmosphere, by means of doors or windows or both may be employed.

(2) Where the combined floor area of the machinery room and adjacent halls or rooms is not less than one (1.0) square foot for each pound of carbon dioxide in the system, no mechanical means of ventilation need be employed, provided:

*a'*. The machinery room and adjacent halls or rooms are connected with the machinery room by means of at least two doors or windows or equivalent wall openings and the sills of such doors, windows or equivalent wall openings are not more than twelve (12) inches above the machinery room floor.

*b'*. The floor level of the adjacent halls or rooms is not more than three (3) feet below the sill of a connecting door, window or equivalent wall opening.

(3) Where a mechanical system of ventilation is employed, the control of such system shall be easily accessible and located outside of the machinery room. The center line of the fan suction or the intake ends of the suction duct shall be within twelve (12) inches of the machinery room floor level.

*Rule* 2104.—*Charging.*

(*a*) Every equipment shall be charged through a connection located on the low pressure side.

## SECTION 220

### Class B Equipment

A "Class B Equipment" is a refrigerating system of less than 30-ton capacity, or containing less than one thousand (1000) pounds of carbon dioxide and over twenty-five (25) pounds of carbon dioxide.

*Rule* 2200.—*Pressure-Imposing Element.*

(*a*) Every equipment, except that which cannot be subjected to a pressure in excess of the maximum allowable working pressure by the action of the pressure-imposing element, shall be provided with one pressure relief valve of at least one-half (½) inch (nominal pipe) size connected between each main discharge stop valve and the pressure-imposing element to relieve excessive pressures, during the period of excess pressure, and discharge into the machinery room. This pressure relief valve shall be so designed as to prevent pressures in the high-pressure side exceeding the maximum allowable working pressure.

(*b*) Every equipment, except that which cannot be subjected to a pressure in excess of the maximum allowable working pressure by the action of the pressure-imposing element, shall have an automatic pressure limiting device, acting directly responsive to pressure, to stop the action of the pressure-imposing element at a pressure not higher than ninety per cent (90%) of the pressure stamped on the relief valve.

*Rule* 2201.—*Liquid Receiver, Condenser and Shell Type Evaporator.*

(*a*) Each liquid receiver, battery of double pipe condenser, shell type condenser and shell type evaporator, which can be isolated, shall each be equipped with a one-half ($\frac{1}{2}$) inch pressure relief valve set to discharge at a pressure not higher than the maximum allowable working pressure, and connected at the highest point to discharge into the machinery room.

(*b*) The low-pressure side of every equipment containing more than five hundred (500) pounds of carbon dioxide, except where the low-pressure side cooling surface is entirely submerged in a liquid, shall be equipped with not less than a one-half ($\frac{1}{2}$) inch pressure relief valve set to discharge at a pressure not higher than the maximum allowable working pressure.

(*c*) On equipments containing less than five hundred (500) pounds of carbon dioxide, a rupture member of not less than one-quarter ($\frac{1}{4}$) inch diameter of orifice may be provided as an alternate to the pressure relief valve. The rupture member shall be designed to rupture at the maximum allowable working pressure, and may be fitted to discharge into the machinery room.

(*d*) In place of each pressure relief valve (Rule 2201 (*a*) and (*b*)), a rupture member may be employed and its discharge side piped to the outside of the building, except that in equipments containing less than five hundred (500) pounds of carbon dioxide, the rupture member may discharge into the machinery room. The rupture member shall be designed to rupture at the maximum allowable working pressure of the vessel to which it is connected.

(*e*) Piping from rupture members may bejoined to a common pipe and thence piped to the outside of the building. No piping from rupture members shall be less than one-half ($\frac{1}{2}$) inch iron pipe size and for each one hundred (100) feet of run the pipe shall increase one pipe size. The free end of any pipe shall be protected by a suitable gooseneck or screen.

(*f*) The rupture members shall be designed to prevent their ruptured parts entering the discharge pipe connections.

(*g*) The rupture orifice for any rupture member shall be not less than one-quarter ($\frac{1}{4}$) inch in diameter and the orifice diameter shall be legibly stamped on the outside of the rupture member body.

(*h*) In place of pressure relief devices mentioned in Rules 2200 (*a*) and 2201 (*a*), (*b*), (*c*), and (*d*), combination devices consisting of a rupture element and a pressure relief valve may be used. These devices shall function at the maximum allowable working pressure. The dimensions of the combination devices shall be the same as specified for relief valves, except that no rupture element in a combination device shall have an orifice less than one-half ($\frac{1}{2}$) inch in diameter. The relief element may discharge into the machinery room. This combination device shall be so designed that pressure is exerted against the relief valve only after the rupture element has functioned.

*Rule* 2202.—*Gauges, Gauge Glasses.*

(*a*) Every equipment shall be provided with pressure gauges; one for indicating pressure in condenser or high-pressure side and one for indicating pressure in evaporator or low-pressure side.

(*b*) Refrigerant level gauge glasses, if used, shall have automatic closing shut-off valves. Such glasses shall be protected by enclosing them in metal casings having longitudinal slots on two opposite sides. The walls of the casing shall be at least one-sixteenth ($\frac{1}{16}$) inch thick and so supported that impacts on the casing are not liable to be transmitted to the glass.

*Rule* 2203.—*Exits, Ventilation.*

(*a*) Each machinery room shall have a direct exit to the open air, or to a room or hall from which gases from machinery room can be excluded by close fitting door.

(*b*) Each machinery room shall be provided with adequate ventilation, as follows:

(1) Where the machinery room floor level is not more than four (4) feet below grade, natural ventilation, to outside atmosphere, by means of doors or windows or both may be employed.

(2) Where the combined floor area of the machinery room and adjacent halls or rooms is not less than one (1.0) square foot for each pound of carbon dioxide in the system, no mechanical means of ventilation need be employed, provided:

*a'*. The machinery room and adjacent halls or rooms are connected with the machinery room by means of at least two doors or windows or equivalent wall openings and the sills of such doors, windows or equivalent wall openings are not more than twelve (12) inches above the machinery room floor.

*b'*. The floor level of the adjacent halls or rooms is not more than three (3) feet below the sill of a connecting door, window or equivalent wall opening.

(3) Where a mechanical system of ventilation is employed, the control of such system shall be easily accessible and located outside of the machinery room. The center line of the fan suction or the intake ends of the suction duct shall be within twelve (12) inches of the machinery room floor level.

(*c*) On systems containing less than five hundred (500) pounds of carbon dioxide and where the combined machinery room floor space and adjacent floor space approximately on the same level, communicable by door, is not less than two hundred (200) square feet floor area, no mechanical means of ventilation need be provided.

*Rule* 2204.—*Charging.*

(*a*) Every equipment shall be charged through a connection located on the low-pressure side.

## SECTION 230

### Class C Equipment

A "Class C Equipment" is a refrigerating system containing twenty-five (25) pounds of carbon dioxide or less.

*Rule* 2300.—*Pressure Limiting Devices*[4] *(Operating Safety Provisions).*

(*a*) Every Class C Equipment shall be provided with at least one rupture member designed to rupture at not higher than the maximum allowable working pressure. The rupture member may discharge into the building space occupied by the equipment.

[4] It is recommended that only such devices which have been designated as suitable for the purpose by a competent disinterested laboratory be used.

(*b*) Every Class C Equipment containing more than fifteen (15) pounds of carbon dioxide and that may be subjected to a pressure in excess of the maximum allowable working pressure by the action of the pressure-imposing element, shall be provided with a pressure-limiting device to stop the action of the pressure-imposing element.

(*c*) Such pressure-limiting device shall be so constructed and connected as to act directly responsive to the discharge pressure. There shall be no valve, except necessary service valves, between this device and the pressure-imposing element.

(*d*) Pressure-limiting devices shall be set, marked and sealed to stop the action of the pressure-imposing element at a pressure not in excess of the maximum allowable working pressure, or shall be so designed that they cannot be adjusted to operate at a greater pressure than the pressure stamped thereon.

*Rule* 2301.—*Electrical Equipment.*

(*a*) All electrical equipment shall be installed in accordance with the National Electrical Code.

## Part III—Sulphur Dioxide Systems

### SECTION 300

#### General Requirements for Sulphur Dioxide Systems

(*a*) Every part of the high-pressure side of the system shall be designed for a working pressure of at least seventy-five (75) pounds per square inch gauge pressure and every part of the low-pressure side shall be designed for a working pressure of at least thirty-five (35) pounds per square inch gauge pressure. A factor of safety of five (5) shall be used in all designs, except for gauges and control mechanism.

(*b*) Every vessel in the system, except connecting piping, shall be marked with the maximum allowable working pressure and shall be tested hydrostatically to one and one-half ($1\frac{1}{2}$) times this pressure marking, such tests to be conducted in accordance with the "Unfired Pressure Vessel Code."

### SECTION 320

#### Class B Equipment

A "Class B Equipment" is a refrigerating system of less than thirty (30) tons' capacity, or containing less than one thousand (1000) pounds and over fifteen (15) pounds of sulphur dioxide.

*Rule* 3200.—*Pressure-Imposing Element.*

(*a*) Every equipment, except that which cannot be subjected to a pressure in excess of the maximum allowable working pressure by the action of the pressure-imposing element, shall be provided with a one-half ($\frac{1}{2}$) inch automatic by-pass connected between each main discharge stop valve and the pressure-imposing element to relieve excessive pressure into the low-pressure side on either side of the main suction stop valve; or, as an alternative, through a one-half ($\frac{1}{2}$) inch (standard pipe) pressure relief valve discharging as specified in Rule 3203. These automatic by-passes or pressure relief valves shall be so designed as to prevent pressures in the high-pressure side exceeding the maximum allowable working pressure.

(*b*) Every equipment, except that which cannot be subjected to a pressure in excess of the maximum allowable working pressure by the action of the pressure-imposing element, shall have an automatic pressure-limiting device, acting directly

responsive to pressure, to stop the action of the pressure-imposing element at a pressure not higher than ninety per cent (90%) of the maximum allowable working pressure.

*Rule* 3201.—*Liquid Receiver, Condenser, Evaporator.*

(*a*) Each liquid receiver, shell type condenser and shell type evaporator, which can be isolated, shall be equipped with a one-half ($\frac{1}{2}$) inch automatic by-pass, set to discharge at a pressure not higher than the maximum allowable working pressure, and connected at the highest point, to discharge to that part of the low-pressure side of the equipment protected by an emergency relief device. As an alternative, a pressure relief valve discharging to the atmosphere may be used. (See Rule 3203.)

*Rule* 3202.—*Fire Emergency Devices.*

(*a*) Every equipment shall be provided with a one-half ($\frac{1}{2}$) inch pressure relief valve connected to the low-pressure side of the equipment, set to relieve excessive pressures in a manner specified in Rule 3203. These devices shall discharge at a pressure not higher than the maximum allowable working pressure.

(*b*) Class B Equipment containing less than one hundred (100) pounds of sulphur dioxide and so designed or provided with devices as to prevent the existence of pressures in any part of the equipment in excess of the maximum allowable working pressure, may be protected by one pressure relief valve.

(*c*) Class B Equipment containing less than one hundred (100) pounds of sulphur dioxide, in which the design of the pressure-imposing element is such that the generation of pressure in excess of the maximum allowable working pressure is impossible, may be protected by fusible plugs on each part of equipment containing sulphur dioxide which can be exposed to heat from an external source, provided the pressure generated at the temperature necessary to fuse the plug is less than the maximum allowable working pressure.

*Rule* 3203.—*Refrigerant Discharge.*

(*a*) The sulphur dioxide passed by the pressure relief devices shall be discharged either to the atmosphere or to a suitable absorbent.

(*b*) When discharged into the atmosphere, the sulphur dioxide shall be conducted by continuous piping to an outlet turned upward and equipped with a suitable diffuser, designed to mix the sulphur dioxide with air. The piping from the pressure relief device to point of final discharge, including the diffuser, shall be such as not to impose a resistance head of more than ten (10) pounds per square inch on the discharge side of the device. Such piping shall be used only for sulphur dioxide discharge.

*Rule* 3204.—*Helmets or Masks.*

(*a*) At least one (1) helmet or mask of approved make, tested in accordance with the requirements of the United States Bureau of Mines for sulphur dioxide, shall be kept in operative condition and available for the immediate use of responsible attendants, who shall be trained in their use, when the refrigerant charge exceeds twenty-five (25) pounds.

(*b*) Helmets or masks shall be kept in a place easily accessible from outside the machinery room.

*Rule* 3205.—*Gauges, Gauge Glasses.*

(*a*) Every equipment containing over one hundred (100) pounds of sulphur dioxide shall be provided with pressure gauges; one for indicating pressure in con-

denser or high-pressure side and one for indicating pressure in evaporator or low-pressure side.

(*b*) Refrigerant level gauge glasses, if used, shall have automatic closing shut-off valves. Such glasses shall be protected by enclosing them in metal casings having longitudinal slots in two opposites sides. The walls of the casing shall be at least one-sixteenth ($\frac{1}{16}$) inch thick and so supported that impacts on the casing are not liable to be transmitted to the glass.

*Rule* 3206.—*Exits, Ventilation.*

(*a*) Each machinery room shall have a direct exit to the open air or to a room or hall from which gases from machinery room can be excluded by a close fitting, self-closing door.

(*b*) Vertical and horizontal openings that permit the passage of gases to other parts of the building, which are not a part of the refrigerating plant, shall be sealed or provided with close-fitting, self-closing doors.

(*c*) Each machinery room shall be provided with means for adequate ventilation to the open air, either direct or by means of a suitable duct or ducts. Where a mechanical system of ventilation is employed, control of such system shall be located outside the machinery room, where it can be quickly and safely reached and the system started in case of an emergency. The mechanical system shall be of sufficient capacity to change the air in the machinery room at least once every five (5) minutes.

*Rule* 3207.—*Charging.*

Every equipment shall be charged through a connection located on the low-pressure side.

## SECTION 330

### Class C Equipment

A Class C Equipment is a refrigerating system containing fifteen (15) pounds of sulphur dioxide or less.

*Rule* 3300.—*Pressure-Limiting Devices*[5] (*Operating Safety Provisions*).

(*a*) Every Class C Equipment containing more than (10) pounds of sulphur dioxide and that may be subjected to a pressure in excess of the maximum allowable working pressure, by the action of the pressure-imposing element, shall be provided with a pressure-limiting device to stop the action of the pressure-imposing element.

(*b*) Such pressure-limiting device shall be so constructed and connected as to act directly responsive to the discharge pressure. There shall be no valve, except necessary service valves, between this device and the pressure-imposing element.

(*c*) Pressure-limiting devices shall be set, marked and sealed to stop the action of the pressure-imposing element at a pressure not in excess of the maximum allowable working pressure, or shall be so designed that they cannot be adjusted to operate at a greater pressure than the pressure stamped thereon.

*Rule* 3301.—*Electrical Equipment.*

(*a*) All electrical equipment shall be installed in accordance with the National Electrical Code.

[5] It is recommended that only such devices which have been designated as suitable for the purpose by a competent disinterested laboratory be used.

## Part IV—Ethyl and Methyl Chloride Systems

### SECTION 400

#### General Requirements for Ethyl and Methyl Chloride Systems

(*a*) Every part of the high-pressure side of the system shall be designed for the following working pressures:

Systems using { Ethyl Chloride, at least 35 lb. per sq. in gauge pressure.
Methyl Chloride, at least 110 lb. per sq. in. gauge pressure.

Every low-pressure side shall be designed for the following working pressure as follows:

Systems using { Ethyl Chloride, at least 25 lb. per sq. in. gauge pressure.
Methyl Chloride, at least 80 lb. per sq. in. gauge pressure.

A factor of safety of five (5) shall be used in all designs, except for gauges and control mechanism.

(*b*) Every vessel in a refrigerating system, except connecting piping, shall be marked with the maximum allowable working pressure and shall be tested hydrostatically to one and one-half ($1\frac{1}{2}$) times this pressure marking. Such tests to be conducted in accordance with the "Unfired Pressure Vessel Code."

### SECTION 420

#### Class B Equipment

A "Class B Equipment" is a refrigerating system of less than thirty (30) tons' capcity, or containing less than one thousand (1000) pounds and over fifteen (15) pounds of refrigerant.

*Rule* 4200.—*Pressure-Imposing Element.*

(*a*) Every equipment, except that which cannot be subjected to a pressure in excess of the maximum allowable working pressure by the action of the pressure-imposing element, shall be provided with a one-half ($\frac{1}{2}$) inch automatic by-pass connected between each main discharge stop valve and the pressure-imposing element to relieve excessive pressure into the low-pressure side on either side of the main suction stop valve; or, as an alternative, through a one-half ($\frac{1}{2}$) inch (standard pipe) pressure relief valve discharging as specified in Rule 4203 (*a*). These pressure relief devices shall be so designed as to prevent pressures in the high-pressure side exceeding the maximum allowable working pressure.

(*b*) Every equipment, except that which cannot be subjected to a pressure in excess of the maximum allowable working pressure by the action of the pressure-imposing element, shall have an automatic pressure-limiting device, acting directly responsible to pressure, to stop the action of the pressure-imposing element at a pressure not higher than ninety per cent (90%) of the pressure stamped on the relief valve.

*Rule* 4201.—*Liquid Receiver, Condenser, Evaporator.*

(*a*) Each liquid receiver, shell type condenser and shell type evaporator, which can be isolated, shall be equipped with a one half ($\frac{1}{2}$) automatic by-pass, set to discharge at a pressure not higher than the maximum allowable working pressure and connected at the highest point, to discharge to the low-pressure side of the equip-

ment. As an alternative, a pressure relief device discharging to the atmosphere may be used. See Rule 4203 (*a*).

*Rule* 4202.—*Fire Emergency Devices.*

(*a*) Every equipment shall be provided with a one-half ($\frac{1}{2}$) inch pressure relief valve connected to the low-pressure side of the equipment, set to relieve excessive pressures in a manner specified in Rule 4203 (*a*), and shall discharge at a pressure not higher than the maximum allowable working pressure.

(*b*) Class B Equipment containing less than 100 pounds of refrigerant and so designed or provided with devices as to prevent the existence of pressures in any part of the equipment in excess of the maximum allowable working pressure may be protected by one pressure relief valve.

*Rule* 4203.—Refrigerant Discharge.

(*a*) When discharged into the atmosphere, the refrigerant shall be conducted by continuous piping to an outlet turned upward and equipped with a suitable diffuser, designed to mix the refrigerant with air. The piping from the pressure relief valve to point of final discharge, including the diffuser, shall be such as not to impose a resistance head of more than ten (10) pounds per square inch on the discharge side of the valve. Such piping shall be used only for refrigerant discharged and be so located that the gases discharged are not liable to become ignited. Outlets from diffusers shall be above the roof of any building within fifty (50) feet.

(*b*) Outlets from atmospheric pressure relief devices shall connect to the discharge pipe specified in Paragraph (*a*) of this rule, or a similar pipe.

(*c*) The fire department, if any, shall make at least an annual inspection of the equipment.

*Rule* 4204.—*Open Flames.*

(*a*) In machinery rooms there shall be no fire, arc light, or flame open to the atmosphere of the room; any device intended to adequately isolate the fire, arc light, or flame shall have been approved by a competent laboratory, and the administrative authority.

*Rule* 4205.—*Electrical Equipment.*

(*a*) All current carrying devices, bus bars, commutators, collector rings, brushes, etc., shall have all live parts in tight enclosures, or be provided with suitable insulating covering.

(*b*) Extensive electrical equipment other than that essential to the ice making or refrigerating plant shall not be permitted in the machinery room.

(*c*) All starting equipment, if placed in the machinery room, including switches, automatic starters, etc., shall be of the oil immersed or vapor-proof type.

(*d*) If standard open type starting equipment is used, it shall be installed in a room separated from the machinery room by vapor-tight incombustible partition. There shall be no door leading directly from this room to the machinery room. Stationary fireproof windows may be installed. A remote control switch or switches of approved vapor-proof type may be installed in the machinery room for operating the starters, etc., located in the switchboard room.

(*e*) Transformers shall be in locked vaults, which, if adjoining the machinery room, shall be separated therefrom preferably by unpierced walls, ceilings and floors of six (6) inches of reinforced concrete or its equivalent. All conduit ends and wall openings for conduits entering the vault shall be sealed. See also, the National Electrical Code.

*Rule* 4206.—*Gauges, Gauge Glasses.*

(*a*) Every equipment shall be provided with pressure gauges; one for indicating pressure in condenser, or high-pressure side, and one for indicating pressure in evaporator, or low-pressure side.

(*b*) Refrigerant level gauge glasses, if used, shall have automatic closing shut-off valves. Such glasses shall be protected by enclosing them in metal casings having longitudinal slots on two opposite sides. The walls of the casing shall be at least one-sixteenth ($\frac{1}{16}$) inch thick and so supported that impacts on the casing are not liable to be transmitted to the glass.

*Rule* 4207.—*Exits, Ventilation.*

(*a*) Each machinery room shall have direct exit to the open air.

(*b*) The machinery room shall be independently provided with means for adequate ventilation to the open air.

(*c*) When a mechanical system of ventilation is employed, the system shall be capable of producing a complete change of air at least once in five (5) minutes. The control of such mechanical system shall be easily accessible and located outside of the machinery room.

(*d*) When windows are used, such windows shall have an area equal to at least one square foot to each five (5) square feet of floor area.

*Rule* 4208.—*Location.*

(*a*) Machines containing between 12 and 100 pounds of refrigerant shall not be installed in any building in congested districts used for any other than refrigerating purposes, unless such building is of fireproof construction, or all parts of the equipment containing the refrigerant are isolated by fireproof enclosures.

(*b*) Machines containing from 100 to 1000 pounds of refrigerant located in congested districts, shall be installed only in fireproof buildings of not more than one story in height, and shall be located on the grade floor, which shall be of unpierced fireproof construction. The machinery room shall be cut off from the rest of the building by unpierced fireproof walls of not less than eight inches of brick or six inches of reinforced concrete. A direct exit to the open air shall be provided. All the refrigerant shall be confined to the machinery room.

*Rule* 4209.—*Charging.*

(*a*) Every equipment shall be charged through a connection located on the low-pressure side.

*Rule* 4210.—*Precautions.*

(*a*) "No Smoking" regulations shall be vigorously enforced.

## SECTION 430

### Class C Equipment

A "Class C Equipment" is a refrigerating system containing fifteen (15) pounds of refrigerant or less.

*Rule* 4300.—*Pressure Limiting Devices*[6] (*Operating Safety Provisions*).

(*a*) Every Class C Equipment containing more than ten (10) pounds of refrigerant that may be subjected to a pressure in excess of the maximum allowable working

[6] It is recommended that only such devices which have been designated as suitable for the purpose by a competent disinterested laboratory be used.

pressure, by the action of the pressure imposing element, shall be provided with a pressure limiting device to stop the action of the pressure imposing element.

(*b*) Such pressure limiting device shall be so constructed and connected as to act directly responsive to the discharge pressure. There shall be no valve, except necessary service valves, between this device and the pressure imposing element.

(*c*) Pressure limiting devices shall be set, marked and sealed to stop the action of the pressure imposing element at a pressure not in excess of the working pressure, or shall be so designed that they cannot be adjusted to operate at a greater pressure than the pressure stamped thereon.

*Rule* 4301.—*Electrical Equipment.*

(*a*) All electrical equipment shall be installed in accordance with the National Electrical Code.

## Part V—Hydro-Carbon Systems

### SECTION 500

#### General Requirements for Hydro-carbon Systems

(*a*) Every part of the high-pressure side of the system shall be designed for the following maximum allowable working pressures:

| | | |
|---|---|---|
| Systems using | Ethane, | at least 1000 lb. per sq. in. gauge pressure. |
| | Propane, | at least 200 lb. per sq. in. gauge pressure. |
| | Freezol, | at least 75 lb. per sq. in. gauge pressure. |
| | Butane, | at least 50 lb. per sq. in. gauge pressure. |

Every part of the low-pressure side shall be designed for the following maximum allowable working pressures:

| | | |
|---|---|---|
| Systems using | Ethane, | at least 500 lb. per sq. in. gauge pressure. |
| | Propane, | at least 150 lb. per sq. in. gauge pressure. |
| | Freezol, | at least 35 lb. per sq. in. gauge pressure. |
| | Butane, | at least 25 lb. per sq. in. gauge pressure. |

A factor of safety of five (5) shall be used in all designs, except for gauges and control mechanism.

(*b*) Every vessel in the system, except connecting piping, shall be marked with the maximum allowable working pressure and shall be tested hydrostatically to one and one-half ($1\frac{1}{2}$) times this pressure marking, such tests to be conducted in accordance with the "Unfired Pressure Vessel Code."

### SECTION 510

#### Class A Equipment

##### ETHANE—PROPANE

A "Class A Equipment" is a refrigerating system of thirty (30) tons capacity or over, or containing one thousand (1000) pounds of refrigerant or over.

*Rule* 5100.—*Pressure Imposing Element.*

(*a*) Every equipment shall be provided with one or more automatic by-passes of proper size, connected between each main discharge stop valve and the pressure-imposing element, to relieve excessive pressures into the low-pressure side, on either

side of the main suction stop valve; or, as an alternative, shall be provided with one or more pressure relief valves of proper size discharging, during the period of excess pressure, to the outside atmosphere. (See Rule 5103 (*a*).) These pressure relief devices shall be so designed as to prevent pressures in the high-pressure side exceeding the maximum allowable working pressure. The proper (nominal pipe) sizes of such devices shall be as follows:

For equipment of from
- 30 to 60 tons capacity, use at least one $\frac{3}{4}$ in. device.
- 60 to 100 tons capacity, use at least one 1 in. device.
- 100 to 175 tons capacity, use at least one $1\frac{1}{4}$ in. device.
- 175 to 250 tons capacity, use at least one $1\frac{1}{2}$ in. device.
- 250 to 450 tons capacity, use at least one 2 in. device.
- 450 to 900 tons capacity, use at least two 2 in. devices.

(*b*) Every equipment shall have an automatic pressure-limiting device, acting directly responsive to pressure, to stop the action of the pressure-imposing element at a pressure not higher than 90% of the pressure stamped on the relief device.

(*c*) A cushioned or other satisfactory check valve shall be placed in the discharge line between the pressure-imposing element and the condenser.

*Rule* 5101.—*Liquid Receiver, Condenser, Evaporator.*

(*a*) Each liquid receiver, shell type condenser and shell type evaporator, which can be isolated, shall be equipped with a one-half ($\frac{1}{2}$) inch automatic by-pass set to discharge at a pressure not higher than the maximum allowable working pressure, and connected at the highest point to discharge to that part of the low-pressure side of the equipment protected by an emergency relief device. As an alternative, a pressure relief valve discharging to the atmosphere may be used. (See Rule 5103 (*a*).)

*Rule* 5102.—*Fire Emergency Devices.*

(*a*) The low-pressure side of every equipment shall be provided with a hand operated relief valve for discharging the refrigerant in case of fire to the atmosphere.

(*b*) The hand operated relief valve shall be located on the outside of the wall of the engine room, or shall be capable of operation therefrom. The valve or operating mechanism shall have a protecting cover.

(*c*) The minimum (standard pipe) sizes of the hand relief valves shall be as follows:

Where the charge is
- 1,000 to 1,800 lb., use one $\frac{3}{4}$ in. valve.
- 1,800 to 3,000 lb., use one 1 in. valve.
- 3,000 to 5,250 lb., use one $1\frac{1}{4}$ in. valve.
- 5,250 to 7,500 lb., use one $1\frac{1}{2}$ in. valve.
- 7,500 to 13,500 lb., use one 2 in. valve.
- 13,500 to 27,000 lb., use one 3 in. valve.

*Rule* 5103.—*Refrigerant Discharge.*

(*a*) When discharged into the atmosphere, the refrigerant shall be conducted by continuous piping to an outlet turned upward and equipped with a suitable diffuser, designed to mix the refrigerant with air. The piping from the hand relief valve to point of final discharge, including the diffuser, shall be such as not to impose a resistance head of more than ten (10) pounds per square inch on the discharge side of the valve. Such piping shall be used only for refrigerant discharge and be so located that the gases discharged are not liable to become ignited. Outlets from diffusers shall be above the roof of any building within fifty (50) feet.

(*b*) Outlets from atmospheric pressure relief devices shall connect to the discharge pipe specified in Paragraph (*a*) of this rule, or a similar pipe.

(*c*) The fire department, if any, shall make at least an annual inspection of the equipment.

*Rule* 5104.—*Open Flames.*

(*a*) In machinery rooms there shall be no fire, arc light or flame open to the atmosphere of the room; any device intended to adequately isolate the fire, arc light or flame shall have been approved by a competent laboratory and the administrative authority.

*Rule* 5105.—*Electrical Equipment.*

(*a*) All current carrying devices, bus bars, commutators, collector rings, brushes, etc., shall have all live parts in tight enclosures, or be provided with suitable insulating covering.

(*b*) Extensive electrical equipment other than that essential to the ice making or refrigerating plant shall not be permitted in the machinery room.

(*c*) All starting equipment, including switches, automatic starters, etc., if placed in the machinery room, shall be of the oil immersed or vapor-proof type.

(*d*) If standard open type starting equipment is used, it shall be installed in a room separated from the machinery room by a vapor-tight incombustible partition. There shall be no door leading directly from this room to the machinery room. Stationary fireproof windows may be installed. A remote control switch or switches of approved vapor-proof type may be installed in the machinery room for operating the starters, etc., located in the switchboard room.

(*e*) Transformers shall be in locked vaults, which, if adjoining the machinery room, shall be separated therefrom preferably by unpierced walls, ceilings and floors of six (6) inches of reinforced concrete or its equivalent. All conduit ends and wall openings for conduits entering the vault shall be sealed. See also the National Electrical Code.

*Rule* 5106.—*Gauges, Gauge Glasses.*

(*a*) Every equipment shall be provided with pressure gauges; one for indicating pressure in condenser, or high-pressure side, and one for indicating pressure in evaporator, or low-pressure side.

(*b*) Refrigerant level gauge glasses, if used, shall have automatic closing shut-off valve. Such glasses shall be protected by enclosing them in metal casings having longitudinal slots on two opposite sides. The walls of the casing shall be at least one-sixteenth ($\frac{1}{16}$) inch thick and so supported that impacts on the casing are not liable to be transmitted to the glass.

*Rule* 5107.—*Exits, Ventilation.*

(*a*) Each machinery room shall have direct exit to the open air.

(*b*) The machinery room shall be independently provided with means for adequate ventilation to the open air.

(*c*) When a mechanical system of ventilation is employed, the system shall be capable of producing a complete change of air at least once in five (5) minutes. The control of such mechanical system shall be easily accessible and located outside of the machinery room.

(*d*) When windows are used, such windows shall have an area equal to at least one square foot to each five (5) square feet of floor area.

*Rule* 5108.—*Location.*

(*a*) Machines shall not be installed in any buildings located in congested districts.

(*b*) Direct expansion shall not be used, unless all parts of the equipment containing refrigerant be confined within fireproof enclosures.

*Rule* 5109.—*Charging.*

(*a*) Every equipment shall be charged through a connection located on the low pressure side.

*Rule* 5110.—*Precautions.*

(*a*) To facilitate the location of leaks, it is recommended that the refrigerant be slightly odorized.

(*b*) "No Smoking" regulations shall be vigorously enforced.

## SECTION 520

### Class B Equipment

ETHANE—PROPANE—FREEZOL—BUTANE

A "Class B Equipment" is a refrigerating system of less than thirty (30) tons capacity, or containing less than one thousand (1000) pounds and over twelve (12) pounds of refrigerant.

*Rule* 5200.—*Pressure-Imposing Element.*

(*a*) Every equipment, except that which cannot be subjected to a pressure in excess of the maximum allowable working pressure by the action of the pressure imposing element, shall be provided with a one-half ($\frac{1}{2}$) inch automatic by-pass connected between each main discharge stop valve and the pressure-imposing element to relieve excessive pressure into the low-pressure side on either side of the main suction stop valve; or, as an alternative, through a one-half ($\frac{1}{2}$) inch (standard pipe) pressure relief valve discharging as specified in Rule 5203 (*a*). These pressure relief devices shall be so designed as to prevent pressures in the high-pressure side exceeding the maximum allowable working pressure.

(*b*) Every equipment, except that which cannot be subjected to a pressure in excess of the maximum allowable working pressure by the action of the pressure-imposing element, shall have an automatic pressure-limiting device, acting directly responsive to pressure, to stop the action of the pressure-imposing element at a pressure not higher than ninety per cent (90%) of the pressure stamped on the relief device.

*Rule* 5201.—*Liquid Receiver, Condenser, Evaporator.*

(*a*) Each liquid receiver, shell type condenser and shell type evaporator, which can be isolated, shall be equipped with a one-half ($\frac{1}{2}$) inch automatic by-pass, set to discharge at a pressure not higher than the maximum allowable working pressure and connected at the highest point, to discharge to the low-pressure side of the equipment. As an alternative, a pressure relief valve discharging to the atmosphere may be used. See Rule 5203 (*a*).

*Rule* 5202.—*Fire Emergency Devices.*

(*a*) Every equipment shall be provided with a one-half ($\frac{1}{2}$) inch pressure relief valve connected to the low-pressure side of the equipment, set to relieve excessive

pressures in a manner specified in Rule 5203 (*a*), and shall discharge at a pressure not higher than the maximum allowable working pressure.

(*b*) Class B Equipment containing less than 100 pounds of refrigerant and so designed or provided with devices as to prevent the existence of pressures in any part of the equipment in excess of the maximum allowable working pressure may be protected by one pressure relief valve.

*Rule* 5203.—*Refrigerant Discharge.*

(*a*) When discharged into the atmosphere, the refrigerant shall be conducted by continuous piping to an outlet turned upward and equipped with a suitable diffuser, designed to mix the refrigerant with air. The piping from the pressure relief valve to point of final discharge, including the diffuser, shall be such as not to impose a resistance head of more than ten (10) pounds per square inch on the discharge side of the valve. Such piping shall be used only for refrigerant discharge and be so located that the gases discharged are not liable to become ignited. Outlets from diffusers shall be above the roof of any building within fifty (50) feet.

(*b*) Outlets from atmospheric pressure relief devices shall connect to the discharge pipe specified in Paragraph (*a*) of this rule, or a similar pipe.

(*c*) The fire department, if any, shall make at least an annual inspection of the equipment.

*Rule* 5204.—*Open Flames.*

(*a*) In machinery rooms there shall be no fire, arc light, or flame open to the atmosphere of the room; any device intended to adequately isolate the fire, arc light, or flame shall have been approved by a competent laboratory and the administrative authority.

*Rule* 5205.—*Electrical Equipment.*

(*a*) All current carrying devices, bus bars, commutators, collector rings, brushes, etc., shall have all live parts in tight enclosures, or be provided with suitable insulating covering.

(*b*) Extensive electrical equipment other than that essential to the ice making or refrigerating plant shall not be permitted in the machinery room.

(*c*) All starting equipment, including switches, automatic starters, etc., if placed in the machinery room, shall be of the oil immersed or vapor-proof type.

(*d*) If standard open type starting equipment is used, it shall be installed in a room separated from the machinery room by a vapor-tight incombustible partition. There shall be no door leading directly from this room to the machinery room. Stationary fireproof windows may be installed. A remote control switch or switches of approved vapor-proof type may be installed in the machinery room for operating the starters, etc., located in the switchboard room.

(*e*) Transformers shall be in locked vaults, which, if adjoining the machinery room, shall be separated therefrom preferably by unpierced walls, ceilings, and floors of six (6) inches of reinforced concrete or its equivalent. All conduit ends and wall openings for conduits entering the vault shall be sealed. See, also, the National Electrical Code.

*Rule* 5206.—*Gauges, Gauge Glasses.*

(*a*) Every equipment shall be provided with pressure gauges; one for indicating pressure in condenser, or high-pressure side, and one for indicating pressure in evaporator, or low-pressure side.

(*b*) Refrigerant level gauge glasses, if used, shall have automatic closing shut-off valves. Such glasses shall be protected by enclosing them in metal casings having longitudinal slots on two opposite sides. The walls of the casing shall be at least one-sixteenth ($\frac{1}{16}$) inch thick and so supported that impacts on the casing are not liable to be transmitted to the glass.

*Rule* 5207.—*Exits, Ventilation.*

(*a*) Each machinery room shall have direct exit to the open air.

(*b*) The machinery room shall be independently provided with means for adequate ventilation to the open air.

(*c*) When a mechanical system of ventilation is employed, the system shall be capable of producing a complete change of air at least once in five (5) minutes. The control of such mechanical system shall be easily accessible and located outside of the machinery room.

(*d*) When windows are used, such windows shall have an area equal to at least one square foot to each five (5) square feet of floor area.

*Rule* 5208.—*Location.*

(*a*) Machines containing between 12 and 100 pounds of refrigerant shall not be installed in any building in congested districts used for any other than refrigerating purposes, unless such building is of fireproof construction, or all parts of the equipment containing the refrigerant are isolated by fireproof enclosures.

(*b*) Machines containing from 100 to 1000 pounds of refrigerant located in congested districts, shall be installed only in fireproof buildings of not more than one story in height, and shall be located on the grade floor, which shall be of unpierced fireproof construction. The machinery room shall be cut off from the rest of the building by unpierced fireproof walls of not less than eight inches of brick or six inches of reinforced concrete. A direct exit to the open air shall be provided. All the refrigerant shall be confined to the machinery room.

*Rule* 5209.—*Charging.*

(*a*) Every equipment shall be charged through a connection located on the low-pressure side.

*Rule* 5210.—*Precautions.*

(*a*) To facilitate the location of leaks, it is recommended that the refrigerant be slightly odorized.

(*b*) "No Smoking" regulations to be vigorously enforced.

## SECTION 530

### Class C Equipment

#### BUTANE—ISOBUTANE—PROPANE

A "Class C Equipment" is a refrigerating system containing twelve (12) pounds of refrigerant or less.

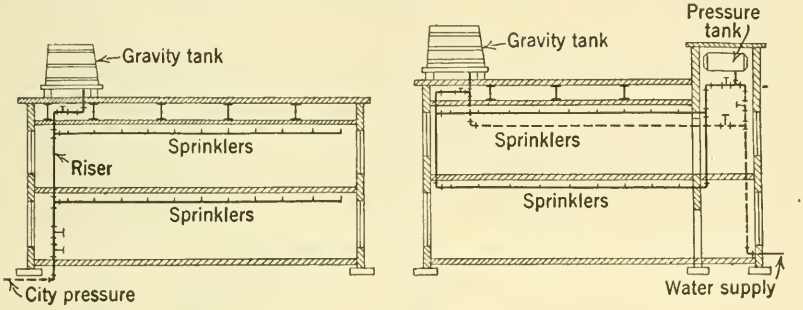
*Rule* 5300.—*Pressure-Limiting Devices*[7] (*Operating Safety Provision*).

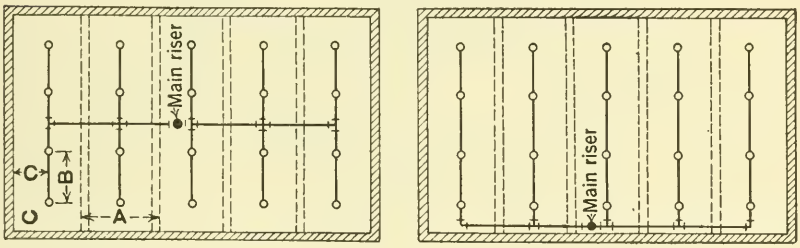
(*a*) Every Class C Equipment containing more than eight (8) pounds of refrigerant that may be subjected to a pressure in excess of the maximum allowable working

[7] It is recommended that only such devices which have been designated as suitable for the purpose by a competent disinterested laboratory be used.

pressure, by the action of the pressure-imposing element, shall be provided with a pressure-limiting device to stop the action of the pressure-imposing element.

(*b*) Such pressure-limiting device shall be so constructed and connected as to act directly responsive to the discharge pressure. There shall be no valve, except necessary service valves, between this device and the pressure-imposing element.



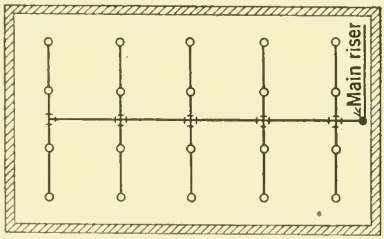




FIG. 382.—Sprinkler Pipe Arrangements.

(*c*) Pressure-limiting devices shall be set, marked and sealed to stop the action of the pressure-imposing element at a pressure not in excess of the working pressure, or shall be so designed that they cannot be adjusted to operate at a greater pressure than the pressure stamped thereon.

*Rule* 5301.—*Electrical Equipment.*

(*a*) All electrical equipment shall be installed in accordance with the National Electrical Code.

**Fire Protection.**—Complete fire protection consists of automatic sprinklers, alarm service, inside protection, including a gravity or pressure water tank and an outside protection which must needs be the city water supply (if under sufficient pressure) or a separate fire pump installation.

**Sprinklers.**—The usual sprinkler head is designed to melt at 165 deg. F., and is designed to be spaced from 8 to 12 ft. apart. For cold storage work it is essential that the piping to the sprinklers be safeguarded against freezing. In such cases a non-freezing (brine) solution can be used or a dry system can be installed using compressed air, in which case (when a sprinkler head melts) the air will pass out through

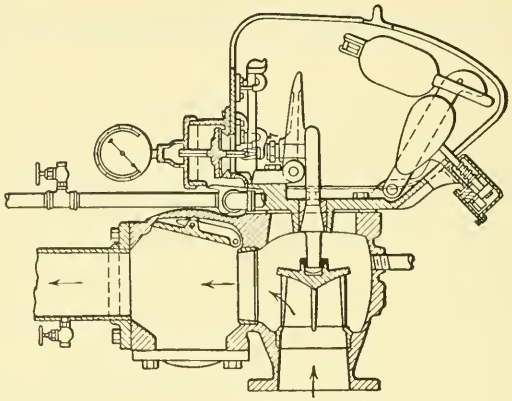


Fig. 383.—Dry Pipe Valve.

the sprinkler head first and will be followed by the water. If a dry pipe sprinkler system is employed, then a calcium chloride reservoir, as shown in Fig. 384, is advised in order to remove the water in the compressed air.

Typical arrangements of piping are shown in Fig. 382. The usual number of sprinklers to be supplied by different sizes of pipe are shown in Table 119.

TABLE 119

| Diameter of Pipe, Inches | Maximum Number of Sprinklers | Diameter of Pipe, Inches | Maximum Number of Sprinklers |
|---|---|---|---|
| $\frac{3}{4}$ | 1 | 3 | 36 |
| 1 | 2 | $3\frac{1}{2}$ | 55 |
| $1\frac{1}{4}$ | 3 | 4 | 80 |
| $1\frac{1}{2}$ | 5 | 5 | 140 |
| 2 | 10 | 6 | 200 |
| $2\frac{1}{2}$ | 20 | | |

**Fire Pumps.**—Every fire protection system should have at least two independent water supplies, one of which must be automatic with a limited water supply and the other must be a connection to the public water works, if under sufficient pressure, or a separate fire pump service.

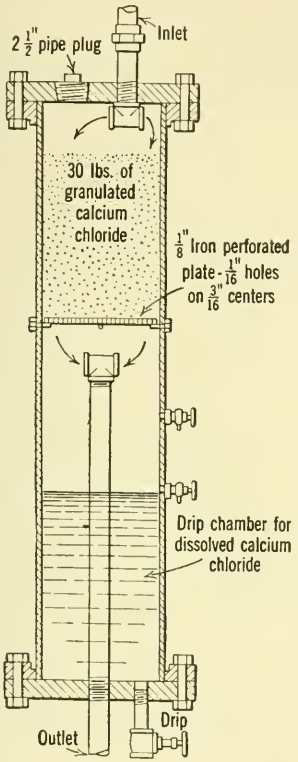



Fig. 384*a*.

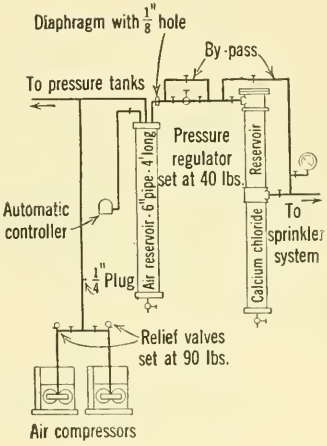



Fig. 384*b*.

Fig. 384.—Calcium Chloride Dryer.

## CHAPTER XX

## COSTS OF REFRIGERATING MACHINERY AND EQUIPMENT

The subject of *costs* of machinery and equipment can not be handled except in a superficial manner. These values are continually changing with the price of pig iron and steel as well as the economic conditions of supply and demand existing from time to time. These purchase prices increased from 1913 to 1919 and 1920, but have since dropped back, and in 1927 they appear to be practically constant.

However, it will be found that the estimates of various manufacturers will vary over a wide range, for as much as 15 to 20 per cent. In equipment such as the common atmospheric condenser, the double pipe condenser (where the specifications call for a definite amount of pipe and fittings), these prices should be in close agreement, but in the case of compressors, etc., the quality of the materials, the total weight and the machine work and care in the assembly may not be comparable at all.

The following, then, are not intended as absolute values but as approximate only, and are to be used simply as a rough estimate of the possible cost.

TABLE 120

VERTICAL ENCLOSED AMMONIA COMPRESSORS (BELT-DRIVEN)

| Size, Inches | Tonnage | R.p.m. | Weight, Pounds | Price, Dollars |
|---|---|---|---|---|
| 3 × 3 (single) | $\frac{3}{4}$ | 255 | 875 | $425 |
| 3 × 3 (double) | $1\frac{1}{2}$ | 258 | 1,075 | 510 |
| 4 × 4 | 3 | 214 | 1,375 | 590 |
| 5 × 5 | 5 | 192 | 2,050 | 720 |
| 6 × 6 | 8 | 170 | 2,625 | 880 |
| $6\frac{1}{2}$ × $6\frac{1}{2}$ | 10 | 170 | 2,975 | 920 |
| $7\frac{1}{2}$ × $7\frac{1}{2}$ | 15 | 165 | 4,300 | 1200 |
| 8 × 8 | 20 | 173 | 5,700 | 1460 |
| 9 × 9 | 25 | 155 | 7,575 | 1920 |
| 10 × 10 | 35 | 158 | 10,800 | 2930 |

Above includes gages, bolts and strainer but does not include oil trap.

TABLE 121

APPROXIMATE DATA

"H.D.A. Belt-driven Compressors"

| Size, Inches | R.p.m. | B. P., Pounds | C. P., Pounds | Refrigeration, Tons | Weight, Pounds | Price, Dollars |
|---|---|---|---|---|---|---|
| 9 ×16 | 120 | 19.58 | 154.5 | 32.0 | 12,500 | 2,400 |
| 10½×18 | 120 | 19.58 | 154.5 | 49.0 | 19,900 | 3,300 |
| 12 ×24 | 100 | 19.58 | 154.5 | 71.0 | 27,100 | 4,200 |
| 13½×26 | 100 | 19.58 | 154.5 | 97.8 | 31,900 | 4.800 |
| "H.D.A. Corliss Engine-driven Compressors" | | | | | | |
| 9 ×13×26 | 120 | 19.58 | 154.5 | 32.0 | 23,000 | 4,800 |
| 10½×15×18 | 120 | 19.58 | 154.5 | 49.0 | 29,000 | 6,700 |
| 12 ×18×24 | 100 | 19.58 | 154.5 | 71.0 | 45,500 | 7,100 |
| 13½×20×26 | 100 | 19.58 | 154.5 | 97.8 | 55,500 | 8,300 |
| "V.S.A. Belt-driven Compressors" | | | | | | |
| 9½×12 | 100 | 19.46 | 154.5 | 23.8 | 15,500 | 3,300 |
| 12½×18 | 90 | 19.46 | 154.5 | 55.9 | 32,500 | 5,000 |
| 13½×20 | 85 | 19.46 | 154.5 | 68.1 | 54,000 | 6,900 |
| 15 ×24 | 80 | 19.46 | 154.5 | 95.3 | 76,000 | 8,800 |
| 16½×28 | 75 | 19.46 | 154.5 | 126.3 | 88,000 | 10,200 |
| "V.S.A. Corliss Engine-driven Compressors" | | | | | | |
| 9½×13×12 | 100 | 19.46 | 154.5 | 23.8 | 20,400 | 4,600 |
| 12½×18×18 | 90 | 19.46 | 154.5 | 55.9 | 37,000 | 7,000 |
| 13½×20×20 | 85 | 19.46 | 154.5 | 68.1 | 60,000 | 11,600 |
| 15 ×22×24 | 80 | 19.46 | 154.5 | 95.3 | 80,000 | 15,200 |
| 16½×24×28 | 75 | 19.46 | 154.5 | 126.3 | 102,000 | 18,800 |
| "V.S.A. Enclosed Belt-driven Compressors" | | | | | | |
| 8 × 8 | 240 | 19.6 | 154.5 | 26.6 | 7,760 | 1,400 |
| 10 ×10 | 225 | 19.6 | 154.5 | 48.8 | 11,660 | 2,000 |
| 12 ×12 | 200 | 19.6 | 154.5 | 72.0 | 22,230 | 3,800 |
| "V.S.A. Semi-enclosed Belt-driven Compressors" | | | | | | |
| 13 ×14 | 180 | 19.0 | 154.5 | 95.3 | 38,500 | 6,300 |
| 15 ×16 | 164 | 19.0 | 154.5 | 132.5 | 53,500 | 8,100 |
| "V.S.A. Enclosed Compressors without Wheel" (for Syn. Motor on Shaft) | | | | | | |
| 8 × 8 | 240 | 19.6 | 154.5 | 26.6 | 5,800 | 1,300 |
| 10 ×10 | 225 | 19.6 | 154.5 | 48.8 | 10,000 | 1,900 |
| 12 ×12 | 200 | 19.6 | 154.5 | 72.0 | 19,700 | 3,600 |
| "V.S.A. Semi-enclosed Compressors without Wheel" (for Syn. Motor on Shaft) | | | | | | |
| 13 ×14 | 180 | 19.0 | 154.5 | 95.3 | 37,000 | 6,100 |
| 15 ×16 | 164 | 19.0 | 154.5 | 132.5 | 51,000 | 8,400 |

TABLE 122

ROLLING MILL TYPE (BELT-DRIVEN)

(Compiled 1920)

| Size of Compressor, Inches | Total Weight, Pounds | Selling Price, Dollars | Maximum R.p.m. | Capacity at 20 and 175 Lb. |
|---|---|---|---|---|
| 6×12 | 7,000 | 1,320 | 100 | 9.65 |
| 7×14 | 8,300 | 1,460 | 100 | 15.32 |
| 8×16 | 10,300 | 1,670 | 100 | 23.65 |
| 9×18 | 13,000 | 1,940 | 100 | 33.61 |
| 10×20 | 18,000 | 2,420 | 90 | 40.40 |
| 11×22 | 22,500 | 2,870 | 90 | 54.0 |
| 12×24 | 24,500 | 3,295 | 90 | 76.2 |
| 13×26 | 32,500 | 3,790 | 80 | 81.5 |
| 14×28 | 36,500 | 4,210 | 80 | 161.6 |
| 15×30 | 42,500 | 4,795 | 80 | 121.8 |
| 16×32 | 48,000 | 5,570 | 80 | 147.0 |
| 17×34 | 52,500 | 6,300 | 75 | 166.2 |
| 18×36 | 68,000 | 7,730 | 75 | 196.8 |
| 19×36 | 75,000 | 8,610 | 75 | 219.8 |
| 19×38 | 78,000 | 9,030 | 75 | 232.0 |
| 20×40 | 88,000 | 10,300 | 70 | 252.0 |
| 21×42 | 104,000 | 10,870 | 70 | 293.5 |
| 22×44 | ..... | ..... | 65 | 312.0 |
| 23×46 | ..... | ..... | 65 | 357.0 |
| 24×48 | ..... | ..... | 65 | 400.0 |

PRICE OF SUCTION AND DISCHARGE VALVES

| Size, Inches | Weight, Pounds | Price, Dollars | Size, Inches | Weight, Pounds | Price, Dollars |
|---|---|---|---|---|---|
| $1\frac{7}{8}$ | 26 | 30 | $1\frac{1}{2}$ | 26 | 30 |
| 4 | 65 | 57 | $3\frac{3}{16}$ | 65 | 50 |
| 6 | 120 | 69 | $5\frac{1}{4}$ | 120 | 69 |

TABLE 123

Ammonia Receivers

| | |
|---|---|
| 5-ton receiver, 7 in. diameter × 9 ft. long | $ 73 |
| 10-ton receiver, 10 in. diameter × 8 ft. long | 104 |
| 25-ton receiver, 16 in. diameter × 8 ft. long | 160 |
| 50-ton receiver, 24 in. diameter × 8 ft. long | 275 |
| 100-ton receiver, 24 in. diameter × 12 ft. long | 325 |

TABLE 124

Ice Making Plants Complete

The mechanical equipment for a 25-ton plant will cost approximately $1100 per ton; for a 50-ton plant, $1000 per ton, and for a 100-ton plant, $900 per ton. These prices would cover machinery, including motors, cranes, tanks, cans, coils, etc., erected in purchaser's building. The approximate prices do not include foundations, insulation, electric wiring, pipe covering and such items, which are generally taken care of by the purchaser and not included in the contract with ice machine manufacturer.

TABLE 125

Horizontal Shell and Tube Brine Coolers

| Capacity, Tons | Surface, Square Feet | Diameter, Inches | Length, Feet | Weight, Pounds | Cost, Dollars |
|---|---|---|---|---|---|
| 4 | 75 | 16½ | 8 | 1,500 | 300 |
| 8 | 150 | 24 | 12 | 3,600 | 550 |
| 25 | 400 | 27 | 16 | 6,900 | 950 |
| 100 | 1600 | 46 | 16 | 16,300 | 200 |

TABLE 126

STEAM-DRIVEN AMMONIA COMPRESSORS

(Compiled 1920)

| Size of Compressor, Inches | Size of Engine, Inches | Frame | Main Bearing, Inches | Weight of Compressor and Engine, Pounds | Price, Total, Dollars |
|---|---|---|---|---|---|
| 6×12 | 8×18 | 00 | 4⅜× 9 | 14,500 | 3,040 |
| 7×14 | 8×18 | 00 | 4⅜× 9 | 16,000 | 3,265 |
| 8×16 | 9 ×18 | 00 | 5⅞× 9 | 17,500 | 3,545 |
| 9×18 | 10×24 | 0 | 6⅜×11 | 25,500 | 4,220 |
| 10×20 | 12×24 | 0 | 6⅜×11 | 30,500 | 4,950 |
| 11×22 | 12×30 | 1 | 7½×14 | 37,000 | 5,515 |
| 12×24 | 13×30 | 1 | 8 ×14 | 41,500 | 6,190 |
| 13×26 | 14×30 | 2 | 8½×16 | 53,000 | 7,090 |
| 14×28 | 16 ×30 | 2 | 9 ×16 | 58,000 | 7,990 |
| 15×30 | 16×36 | 2 | 9½×16 | 64,000 | 8,440 |
| 16×32 | 18×36 | 3 | 10 ×20 | 74,000 | 9,230 |
| 17×34 | 20×36 | 3 | 11× 20 | 78,000 | 9,965 |
| 18×36 | 20×42 | 4 | 11 ×24 | 108,000 | 12,100 |
| 19×36 | 22×42 | 4 | 12 ×24 | 116,000 | 12,940 |
| 19×38 | 22×42 | 4 | 12 ×24 | 120,000 | 13,450 |
| 20×40 | 24×42 | 4 | 13 ×24 | 128,000 | 14,850 |
| 21×42 | 24×48 | 5 | 13 ×28 | 148,000 | 16,660 |
| 22×44 | 26×48 | 5 | 13 ×28 | 152,000 | 17,665 |
| 23×46 | 26×48 | 5 | 14 ×28 | 168,000 | 19,120 |
| 24×48 | 28×48 | 5 | 14 ×28 | 186,000 | 21,380 |

125–150 lb. steam pressure.

TABLE 127

BELT-DRIVEN AMMONIA COMPRESSORS

(Compiled 1920)

*Duplex Belt Driven*

| Size, Inches | Band Wheel | | | Total Weight, Pounds | Price, Dollars |
|---|---|---|---|---|---|
| | Diameter, feet | Face, inches | Weight, pounds | | |
| 2– 6×12 | 7 | 8 | 4,500 | 13,500 | 2,475 |
| 2– 7×14 | 8 | 11 | 5,000 | 15,500 | 2,735 |
| 2– 8×16 | 9 | 15 | 7,000 | 19,000 | 3,400 |
| 2– 9×18 | 9 | 21 | 9,500 | 25,500 | 3,710 |
| 2–10×20 | 10 | 26 | 11,000 | 34,000 | 4,500 |
| 2–11×22 | 10 | 32 | 14,000 | 42,000 | 5,290 |
| 2–12×24 | 14 | 34 | 16,500 | 46,000 | 5,850 |

*Duplex Rope-driven Heavy Duty*

| | | | | | |
|---|---|---|---|---|---|
| 2–13×26 | 16 | | 14,000 | 56,000 | 7,205 |
| 2–14×28 | 18 | | 17,000 | 64,000 | 7,950 |
| 2–15×30 | 20 | | 19,500 | 74,000 | 9,340 |
| 2–16×32 | 20 | | 23,000 | 85,000 | 10,500 |
| 2–17×34 | 22 | | 29,000 | 94,000 | 11,770 |
| 2–18×36 | 22 | | 35,000 | 122,000 | 13,930 |

TABLE 128

Double Pipe Ammonia Condenser

| Tons | Length Overall, Inches | Number of Section | Number of Pipes per Section | Total Length of Pipes, Inches | Weight, Pounds | Price, Dollars |
|---|---|---|---|---|---|---|
| 3 | 12 | 1 | 6 | 72 | 1,200 | 80 |
| 4 | 20 | 1 | 4 | 80 | 1,650 | 75 |
| 5 | 20 | 1 | 5 | 100 | 1,800 | 90 |
| 6 | 20 | 1 | 6 | 120 | 2,000 | 105 |
| 7 | 20 | 1 | 7 | 140 | 2,200 | 135 |
| 8 | 20 | 1 | 8 | 160 | 2,400 | 155 |
| 10 | 20 | 1 | 10 | 200 | 2,700 | 175 |
| 12 | 20 | 1 | 12 | 240 | 3,400 | 230 |
| 16 | 20 | 2 | 8 | 320 | 4,600 | 300 |
| 18 | 20 | 2 | 9 | 360 | 5,600 | 310 |
| 20 | 20 | 2 | 10 | 400 | 6,200 | 345 |
| 22 | 20 | 2 | 11 | 440 | 6,700 | 420 |
| 25 | 20 | 2 | 12 | 480 | 6,900 | 455 |
| 30 | 20 | 4 | 8 | 560 | 7,900 | 600 |
| 40 | 20 | 4 | 10 | 800 | 11,800 | 695 |
| 50 | 20 | 4 | 12 | 960 | 13,500 | 910 |
| 60 | 20 | 5 | 12 | 1200 | 15,700 | 1090 |
| 70 | 20 | 6 | 12 | 1440 | 19,000 | 1360 |
| 80 | 20 | 8 | 10 | 1600 | 23,000 | 1400 |
| 90 | 20 | 8 | 11 | 1760 | 24,250 | 1610 |
| 100 | 20 | 8 | 12 | 1960 | 25,500 | 1820 |
| 120 | 20 | 10 | 12 | 2400 | 27,000 | 2280 |
| 130 | 20 | 12 | 11 | 2640 | 29,000 | 2530 |
| 150 | 20 | 13 | 12 | 3100 | 36,400 | 2760 |

Price includes stands, stop valve on inlet, valves at each end.

TABLE 129

AMMONIA SEPARATORS

Horizontal $NH_3$ Separators

| Size of Pipes, Inches | For Compressor, Inches | | Diameter, Inches | Length, Inches | Thickness of Shell, Inches | Thickness of Heads, Inches | Weight, Pounds | Price, Dollars |
|---|---|---|---|---|---|---|---|---|
| | Simple | Duplex | | | | | | |
| $1\frac{1}{2}$ | 6×12 | ..... | 8 | 24 | $\frac{9}{32}$ | $\frac{3}{8}$ | 100 | 50 |
| $2\frac{1}{2}$ | 9×18 | ..... | 10 | 30 | $\frac{5}{16}$ | $\frac{1}{2}$ | 160 | 75 |
| 3 | 12×24 | ..... | 10 | 30 | $\frac{5}{16}$ | $\frac{1}{2}$ | 160 | 78 |
| 4 | 15×30 | 12×24 | 12 | 30 | $\frac{11}{32}$ | $\frac{9}{16}$ | 240 | 97 |
| 5 | 19×38 | ..... | 16 | 30 | $\frac{3}{8}$ | $\frac{3}{4}$ | 350 | 137 |
| 7 | 23×46 | 19×36 | 20 | 36 | $\frac{1}{2}$ | $\frac{15}{16}$ | 700 | 195 |
| 8 | ..... | 22×44 | 20 | 36 | $\frac{1}{2}$ | $\frac{15}{16}$ | 700 | 230 |

TABLE 130

COST OF ABSORPTION MACHINES

| Description of Material | 25-ton List, Dollars | 50-ton List, Dollars | 75-ton List, Dollars | 100-ton List, Dollars |
|---|---|---|---|---|
| Generator | 2,943 | 3,878 | 5,974 | 6,898 |
| Rectifier (double pipe) | 498 | 836 | 1,940 | 1,991 |
| Exchanger shell and coil | 1,141 | 2,294 | 3,052 | 4,612 |
| Weak liquor cooler (double pipe) | 361 | 580 | 1,095 | 1,672 |
| Condenser (Spiro Flo) | 1,380 | 2,117 | 3,268 | 4,234 |
| Absorber, tubular | 2,500 | 4,900 | 5,900 | 8,310 |
| Aqua pump | 1,300 | 2,060 | 2,887 | 2,887 |
| High and low side connections | 1,175 | 1,960 | 2,170 | 2,650 |
| Receivers aqua | 238 | 255 | 321 | 402 |
| Receivers anhydrous | 200 | 224 | 250 | 320 |
| To insulate generator | 140 | 184 | 233 | 270 |
| To insulate exchanger | 41 | 78 | 89 | 156 |
| Steam connections | 400 | 600 | 900 | 1,200 |
| Water connections | 600 | 900 | 1,200 | 1,500 |
| Ammonia charge (anhydrous) | 140 | 215 | 342 | 407 |
| Ammonia charge (aqua) | 540 | 950 | 1,510 | 1,920 |
| | 13,597 | 22,031 | 31,131 | 39,429 |
| Less 50 and 11 per cent to get selling price | 6,020 | 9,800 | 13,900 | 17,500 |

25-TON ABSORPTION MACHINE CONSISTS OF THE FOLLOWING PARTS

*Generator:* 29 in. diameter by 16 ft. 0 in., containing 349 sq. ft. surface in coils. There will be 7 coils made of 2-in. pipe and return bends.

*Rectifier:* 1 coil, 5 runs made of $1\frac{1}{2}$-in. and 3-in. pipe, of horizontal type, containing 37 sq. ft. of surface.

*Exchanger:* Of shell and coil type, 1 shell 18 in. diameter by 7 ft. 0 in., containing 187 sq. ft. of surface in the 4 1-in. helical coils.

*Weak Liquor Cooler:* 1 coil, 5 runs made of 2-in. and 3-in. pipe, containing 60 sq. ft. of surface.

*Condenser:* 1 vertical tubular spira-flo condenser, $26\frac{1}{2}$ in. diameter by 8 ft. $4\frac{1}{4}$ in., containing 315 sq. ft. surface in the 72 2-in. No. 12 gage seamless steel tubes.

*Absorber:* 1 tubular absorber, 35 in. diameter by 14 ft. 0 in., containing 706 sq. ft. surface in the 100 2-in. No. 12 gage seamless steel tubes.

*Aqua Pump:* 1 duplex steam driven aqua pump, 8 by $3\frac{3}{4}$ by 12, capable of delivering 0.573 gal. per stroke.

*High and Low Side Connections:* Necessary to connect the various pieces of apparatus.

*Receiver:* 1 12 in. by 4 ft. 0 in. anhydrous and one 24 in. by 4 ft. 0 in. aqua receiver.

*Insulation:* Necessary magnesia black, plaster and canvas for generator and exchanger shells.

*Water Connections:* Necessary pipes and fittings for water connections on machine.

*Steam and Exhaust Connections:* Necessary pipe and fittings for steam and exhaust connections on machine.

*Aqua and Anhydrous Charge:* Necessary aqua and anhydrous to charge machine.

50-TON ABSORPTION MACHINE CONSISTS OF THE FOLLOWING PARTS

*Generator:* 36 in. diameter by 18 ft. 0 in., containing 654 sq. ft. surface in coils. There will be 9 coils made of 2-in. pipe and return bends.

*Rectifier:* 1 coil, 6 runs made of 2-in. and 4-in. pipe, of vertical type. Containing 76 sq. ft. surface.

*Exchanger:* of shell and coil type, 1 shell, 24 in. diameter by 10 ft. 0 in., containing 405 sq. ft. of surface in the 4 $1\frac{1}{4}$-in. helical coils.

*Weak Liquor Cooler:* 1 coil, 8 runs made of 2-in. and 3-in. pipe, containing 96 sq. ft. of surface.

*Condenser:* 1 vertical tubular spira-flo condenser, 30 in. diameter by 14 ft. $0\frac{1}{4}$ in., containing 660 sq. ft. of surface in the 90 2-in. No. 12 gage seamless steel tubes.

*Absorber:* One Tubular absorber 48 in. diameter by 14 ft., containing 1680 sq. ft. surface in the 238 2 in. No. 12 gage seamless steel tubes.

*Aqua Pump:* One simplex steam driven aqua pump, 12 by 6 by 12, capable of delivering 1.46 gal. per stroke.

*High and Low Side Connections:* Necessary to connect up various pieces of apparatus.

*Receivers:* 1 12 in. by 8 ft. 0 in. anhydrous and one 24 in. by 8 in. aqua receiver.

*Insulation:* Necessary magnesia black, plaster and canvas for generator and exchanger shells.

*Water Connections:* Necessary pipe and fittings for water connections on machine.

*Steam and Exhaust Connections:* Necessary pipe and fittings for steam and exhaust connection on machine.

*Aqua and Anhydrous Charge:* Necessary aqua and anhydrous to charge machine.

### 75-ton Absorption Machine Consists of the Following Parts

*Generator:* 45 in. diameter by 18 ft. 0 in., containing 1072 sq. ft. surface in coils. There will be 11 coils made of 2-in. pipe and return bends.

*Rectifier:* 2 coils, 6 runs each made of 2-in. and 4-in. pipe, of vertical type, containing 122 sq. ft. of surface.

*Exchanger:* of shell and coil type, 1 shell 29 in. diameter by 9 ft. 0 in., containing 593 sq. ft. surface in the 5 $1\frac{1}{4}$-in. helical coils.

*Weak Liquor Cooler:* 2 coils, 6 runs each, made of 2-in. and 3-in. pipe, containing 144 sq. ft. surface.

*Condenser:* 1 vertical tubular spira-flo condenser, consisting of 2 shells, $26\frac{1}{2}$ in. diameter by 13 ft. $1\frac{1}{2}$ in., containing 495 sq. ft. surface in each shell. Each shell has 72 2-in. No. 12 gage seamless steel tubes.

*Absorber:* 1 tubular absorber, 54 in. diameter by 16 ft. 0 in., containing 2160 sq. ft. of surface in the 268 2-in. No. 12 gage seamless steel tubes.

*Aqua Pump:* 1 simplex steam driven aqua pump, 14 by 7 by 16, capable of delivering 2.66 gal. per stronke.

*High and Low Side Connections:* Necessary to connect various pieces of apparatus.

*Receivers:* 1 12 in. by 12 ft. 0 in. anhydrous and 1 30 in. by 6 ft. 0 in. aqua receiver.

*Insulation:* Necessary magnesia black, plaster and canvas for generator and exchanger shells.

*Water Connections:* Necessary pipe and fittings for water connections on machine.

*Steam and Exhaust Piping:* Necessary pipe and fittings for steam and exhaust connections for machine.

*Aqua and Anhydrous:* Necessary aqua and anhydrous to charge machine.

### 100-ton Absorption Machine Consists of the Following Parts

*Generator:* 45 in. diameter by 21 ft. 0 in., containing 1252 sq. ft. of surface in the coils. There will be 11 coils made of 2-in. pipe return bends.

*Rectifier:* 2 coils, 6 runs each, made of 2-in. and 4-in. pipe, of vertical type, containing 152 sq. ft. surface.

*Exchanger:* of shell and coil type, 2 shells 24 in. diameter by 10 ft. 0 in., containing a total of 810 sq. ft. of surface in the 4 $1\frac{1}{4}$-in. helical coils in each shell.

*Weak Liquor Cooler:* 2 coils, 10 runs each, made of 2-in. and 3-in. pipe, containing 240 sq. ft. surface.

*Condenser:* 1 vertical tubular spira-flo condenser consisting of 2 shells 30 in. diameter by 14 ft. $0\frac{1}{4}$ in., containing 660 sq. ft. surface in each shell. Each shell will have 90 2-in. No. 12 gage seamless steel tubes.

*Absorber:* 1 tubular absorber, 63 in. diameter by 16 ft. 0 in., containing 3159 sq. ft. surface in the 392 2-in. No. 12 gage seamless steel tubes.

*Aqua Pump:* 1 simplex steam driven aqua pump, 14 by 7 by 16, capable of delivering 2.66 gal. per stroke.

*High and Low Side Connections:* Necessary to connect various pieces of apparatus.

*Receiver:* 1 18 in. by 7 ft. 6 in. anhydrous and 1 36 in. by 6 ft. 0 in. aqua receiver.

*Insulation:* Necessary magnesia black, plaster and canvas for generator and exchanger shells.

*Water Connections:* Necessary pipe and fittings for water connections on machine.

*Steam and Exhaust Piping:* Necessary pipe and fittings for steam and exhaust connections for machine.

*Aqua and Anhydrous:* Necessary Aqua and anhydrous to charge machine.

The nearest to a 75-ton cooler we have is one that is rated at 79 tons, having a shell $36\frac{3}{4}$ in. diameter by 13 ft. $1\frac{1}{2}$ in., containing 1025 sq. ft. of surface in the 149 2-in. No. 12 gage seamless steel tube. The selling price of this will be $1255.00.

The nearest to 100-ton cooler we have is one rated at 106 tons, having a shell $46\frac{1}{4}$ in. diameter by 11 ft. $1\frac{1}{2}$ in., containing 1375 sq. ft. surface in the 236 2-in. No. 12 gage seamless steel tubes. Selling price of this will be $1640.00.

These prices are for condensers complete with distributors, cast iron columns, liquid level gage glass, but does not include any valves.

TABLE 131

COOLERS AND CONDENSERS FOR $CO_2$

Horizontal Shell and Tube Brine Cooler

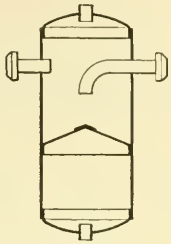
| Tons | Diameter | | Length | | Weight, Pounds | Price, Dollars |
|---|---|---|---|---|---|---|
| | Ft. | In. | Ft. | In. | | |
| 8 | 3 | 6 | 13 | 6 | 4800 | 750 |
| 10 | 3 | 6 | 13 | 6 | 5200 | 800 |
| 12 | 3 | 0 | 20 | 0 | 7000 | 900 |
| 16 | 3 | 6 | 20 | 0 | 8500 | 1025 |

Shell Type Brine Coolers and Condensers—Coil of $\frac{3}{4}$ Ex. Hy. Pipe

3-ton Double Coil Type, 5 ft. 6 in. long, 14 in. diameter, $90
4-ton Double Coil Type, 6 ft. 6 in. long, 14 in. diameter, 130
5-ton Triple Coil Type, 5 ft. 0 in. long, 18 in. diameter, 165
6-ton Triple Coil Type, 6 ft. 0 in. long, 18 in. diameter, 210

TABLE 132

AMMONIA SEPARATOR OR OIL TRAP

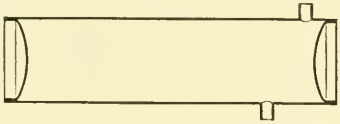


| Size of Compressor, Inches | Size of Pipe Connections, Inches | Diameter, Inches | Length, Inches | Weight, Pounds | List Price, Dollars |
|---|---|---|---|---|---|
| 4×6 | 1 | 6 | 18 | 70 | 34.40 |
| 5×6 | $1\frac{1}{4}$ | 6 | 18 | 75 | 36.10 |
| 6×8 | $1\frac{1}{2}$ | 8 | 24 | 90 | 46.10 |
| 7×8 | $1\frac{1}{2}$ | 8 | 24 | 100 | 47.70 |
| 8×9<br>9×9 | 2 | 8 | 24 | 100 | 47.70 |

Price covers standard separator only, as shown, no valves nor companion flanges included. All required valves, companion flanges, fittings, etc. are included in standard high pressure connections.

HORIZONTAL LIQUID RECEIVERS

(Compiled 1920)



| Size of Compressor, Inches | Diameter, Inches | Length | | Inlet and Outlet, Inches | Weight, Pounds | List Price, Dollars |
|---|---|---|---|---|---|---|
| | | Ft. | In. | | | |
| 4×6 | 8 | 4 | 6 | $\frac{1}{2}$ | 175 | 40.00 |
| 5×6 | 8 | 6 | 0 | $\frac{1}{2}$ | 200 | 43.30 |
| 6×8 | 8 | 10 | 6 | $\frac{3}{4}$ | 325 | 53.30 |
| 7×8 | 8 | 14 | 0 | $\frac{3}{4}$ | 400 | 62.20 |
| 8×9<br>9×9 | 8 | 14 | 0 | 1 | 400 | 62.20 |

Price covers standard receiver only as shown. No valves, companion flanges nor stands are included. All required valves, companion flanges and fittings are included in standard high-pressure connections.

TABLE 133

$CO_2$ Compressors

| Tons | Bore, Inches | Stroke | R.p.m. | Diameter of Wheel, Inches | Weight of Wheel, Pounds | Total Weight, Pounds | Required Horse Power | Price, Dollars |
|---|---|---|---|---|---|---|---|---|
| 3 | $2\frac{1}{4}$ | 6 | 120 | 36 | 1,000 | 2,100 | $7\frac{1}{2}$ | 600 |
| 4 | $2\frac{1}{4}$ | 6 | 135 | 36 | 1,000 | 2,100 | 10 | 600 |
| 5 | $2\frac{1}{2}$ | 9 | 95 | 36 and 60 | 1,700 | 3,800 | 10 | 720 |
| 6 | $2\frac{3}{4}$ | 9 | 110 | 36 and 60 | 1,700 | 3,800 | 15 | 720 |
| 7 | $2\frac{7}{8}$ | 10 | 100 | 36 and 60 | 2,200 | 4,300 | 20 | 725 |
| 8 | $2\frac{7}{8}$ | 10 | 110 | 36 and 60 | 2,200 | 4,300 | 20 | 725 |
| 10 | 3 | $11\frac{1}{2}$ | 90 | 82 | 2,600 | 6,200 | 20 | 1115 |
| 12 | $3\frac{1}{4}$ | $11\frac{1}{2}$ | 90 | 82 | 3,000 | 6,200 | 25 | 1115 |
| 15 | $3\frac{1}{2}$ | 14 | 90 | 82 | 3,000 | 9,500 | 25 | 1600 |
| 18 | $3\frac{5}{8}$ | 14 | 85 | 84 | 4,300 | 10,500 | 30 | 1600 |
| 20 | $3\frac{3}{4}$ | 14 | 95 | 84 | 4,500 | 11,000 | 35 | 1600 |
| 22 | $3\frac{7}{8}$ | 14 | 95 | 84 | 4,500 | 11,000 | 40 | 1600 |
| 25 | $4\frac{1}{4}$ | 16 | 80 | 96 | 6,000 | 14,500 | 40 | 1890 |
| $27\frac{1}{2}$ | | | 90 | | | | | |
| 30 | $4\frac{3}{8}$ | 16 | 90 | 96 | 6,500 | 15,000 | 50 | 1890 |
| 40 | 5 | 20 | 70 | 9 | 8,000 | 20,000 | 60 | 2290 |
| 50 | $5\frac{1}{2}$ | 20 | 70 | 9 | 9,000 | 21,000 | 75 | 2440 |
| 60 | $5\frac{3}{4}$ | 20 | 75 | 12 | 10,000 | 22,000 | 90 | 2575 |
| 70 | $6\frac{1}{4}$ | 24 | 85 | 12 | 11,500 | 25,000 | 100 | 3320 |
| 80 | $6\frac{1}{2}$ | 24 | 65 | 14 | 14,000 | 27,500 | 110 | 3730 |
| 90 | $6\frac{3}{4}$ | 24 | 65 | 14 | 15,000 | 28,500 | 120 | 3875 |
| 100 | 7 | 24 | 65 | 14 | 16,000 | 29,800 | 135 | 4000 |
| 120 | $7\frac{3}{4}$ | 28 | 60 | 16 | 18,000 | 48,000 | 150 | 5000 |
| 130 | 8 | 28 | 60 | 18 | 20,000 | 49,000 | 200 | 3500 |
| 150 | $8\frac{1}{2}$ | 28 | 60 | 18 | 25,000 | 49,000 | 275 | 5500 |

TABLE 134

Triple Pass Shell Type Galvanized Drinking Water Cooler

For $CO_2$

| | |
|---|---|
| 1 15-ton drinking water cooler, 9 ft. 0 in. long, $22\frac{1}{4}$ in. diameter, weight 4200 lb. | $800 |
| Insulation: 4 vertical partitions with gear cork | 100 |
| 1 12-ton, 8 ft. 0 in. long by $22\frac{1}{4}$ in. diameter, weight 3800 lb. | 725 |
| Insulation | 95 |
| 1 10 ton, 7 ft. 6 in. long by $22\frac{1}{4}$ in. diameter, weight 3200 lb. | 650 |
| Insulation | 90 |
| 1 8 ton, 6 ft. 0 in. long, $22\frac{1}{4}$ in. diameter, weight 2800 lb. | 575 |
| Insulation | 85 |

TABLE 135

DOUBLE PIPE BRINE COOLERS, 1¼-IN. AND 2-IN. PIPE, 10 FT. AND 20 FT. LONG OVERALL SPECIFICATION, WEIGHTS AND PRICES

| Pipes High | Effective Surface | Effective Surface When Submerged | Height over Top Pipe, Inches | Height Overall, Inches | Full Weight Pipe | | | Extra Heavy Pipe | | | Add for Soldered Joints |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | Weight, pounds | Black | | Weight, pounds | Black | | |
| | | | | | | Steel | Wrought iron | | Steel | Wrought iron | |
| 10 Ft. Coils | | | | | | | | | | | |
| 4 | 15.0 | 19.2 | 24 | 33 | 490 | $ 98 | $112 | 590 | $108 | $126 | $14 |
| 6 | 19.5 | 27.8 | 32 | 39 | 710 | 146 | 164 | 840 | 156 | 182 | 20 |
| 8 | 26.0 | 38.2 | 40 | 47 | 930 | 190 | 214 | 1080 | 202 | 236 | 26 |
| 10 | 32.5 | 48.0 | 48 | 63 | 1150 | 230 | 262 | 1340 | 246 | 288 | 32 |
| 12 | 39.0 | 57.6 | 56 | 71 | 1370 | 266 | 306 | 1590 | 288 | 338 | 38 |
| 14 | 45.5 | 67.2 | 64 | 79 | 1590 | 298 | 344 | 1840 | 326 | 386 | 44 |
| 16 | 51.5 | 77.0 | 72 | 87 | 1810 | 328 | 380 | 2090 | 362 | 432 | 50 |
| 18 | 58.2 | 86.5 | 80 | 95 | 2030 | 356 | 414 | 2340 | 396 | 476 | 56 |
| 20 | 64.9 | 96.0 | 88 | 113 | 2250 | 382 | 446 | 2590 | 428 | 516 | 62 |
| 20 Ft. Coils | | | | | | | | | | | |
| 4 | 30.4 | 44.4 | 24 | 33 | 740 | 130 | 158 | 860 | 150 | 196 | 14 |
| 6 | 45.6 | 66.6 | 32 | 39 | 1080 | 186 | 226 | 1255 | 214 | 262 | 20 |
| 8 | 61.0 | 89.0 | 40 | 47 | 1420 | 236 | 290 | 1650 | 274 | 342 | 26 |
| 10 | 76.5 | 111.0 | 48 | 63 | 1760 | 286 | 354 | 2045 | 330 | 414 | 32 |
| 12 | 91.0 | 133.2 | 56 | 71 | 2100 | 330 | 414 | 2440 | 382 | 482 | 38 |
| 14 | 106.5 | 155.5 | 64 | 79 | 2440 | 370 | 470 | 2835 | 430 | 546 | 44 |

Gas and liquid valves and 2 pr. flanges for brine connections included with each coil. No headers included.

TABLE 136

BRINE ROOM PIPING. BLACK PIPE

Weight and List Price per Foot of Total Pipe in Coil

| Length of coil —A=..... | 10 Ft. | | 20 Ft. | | 30 Ft. | | 40 Ft. | | 60 Ft. | | 80 Ft. | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Weight, pounds | List, dollars | Weight, pounds | List, dollars | Weight, pounds | List, dollars | Weight, pounds | List, dollars | Weight, pounds | List, dollars | Weight, pounds | List, dollars |
| 2-in. E. H.-W. I. Lap Weld..... | 5.1 | 0.98 | 5.1 | 0.96 | 5.1 | 0.94 | 5.3 | 0.92 | 5.2 | 0.90 | 5.2 | 0.90 |
| 2-in. E. H.-Stl. Butt Weld..... | 5.1 | .60 | 5.1 | .58 | 5.1 | .56 | 5.3 | .54 | 5.2 | .52 | 5.2 | .52 |
| 2-in. F. W.-W. I. Lap Weld..... | 3.7 | .78 | 3.7 | .76 | 3.7 | .74 | 4.0 | .72 | 3.9 | .70 | 3.8 | .70 |
| 2-in. F. W.-Stl. Butt Weld..... | 3.7 | .48 | 3.7 | .46 | 3.7 | .44 | 4.0 | .42 | 3.9 | .40 | 3.8 | .40 |
| 1¼-in. E. H.-W. I. Butt Weld... | 3.2 | .56 | 3.1 | .54 | 3.1 | .52 | 3.5 | .52 | 3.4 | .52 | 3.3 | .52 |
| 1¼-in. E. H.-Stl. Butt Weld..... | 3.2 | .38 | 3.1 | .36 | 3.1 | .34 | 3.5 | .34 | 3.4 | .34 | 3.3 | .34 |
| 1¼-in. F. W.-W. I. Butt Weld... | 2.5 | .46 | 2.3 | .44 | 2.3 | .42 | 2.8 | .42 | 2.7 | .42 | 2.6 | .42 |
| 1¼-in. F. W.-Stl. Butt Weld..... | 2.5 | .30 | 2.3 | .28 | 2.3 | .26 | 2.8 | .26 | 2.7 | .26 | 2.6 | .26 |
| | 1.5 ft. and less | | 2.5 ft. | | 3.5 ft. | | 4.5 ft. | | 5.5 ft. | | 6.5 ft. | |
| 2-in. E. H.-W. I. Lap Weld..... | 6.0 | 1.50 | 5.7 | 1.26 | 5.6 | 1.18 | 5.5 | 1.16 | 5.4 | 1.14 | 5.4 | 1.14 |
| 2-in. E. H.-Stl. Butt Weld..... | 6.0 | 1.16 | 5.7 | .94 | 5.6 | .86 | 5.5 | .84 | 5.4 | .82 | 5.4 | .82 |
| 2-in. F. W.-W. I. Lap Weld..... | ..... | ..... | 4.4 | 1.08 | 4.2 | 1.02 | 4.1 | 1.00 | 4.0 | .98 | 4.0 | .98 |
| 2-in. F. W.-Stl. Butt Weld..... | ..... | ..... | 4.4 | .80 | 4.2 | .74 | 4.1 | .72 | 4.0 | .68 | 4.0 | .68 |
| 1¼-in. E. H.-W. I. Butt Weld... | 4.0 | .76 | 3.6 | .74 | 3.5 | .72 | 3.5 | .70 | 3.4 | .66 | 3.3 | .66 |
| 1¼-in. E. H.-Stl. Butt Weld..... | 4.0 | .60 | 3.6 | .56 | 3.5 | .54 | 3.5 | .52 | 3.4 | .50 | 3.3 | .50 |
| 1¼-in. F. W.-W. I. Butt Weld... | 3.3 | .70 | 3.0 | .66 | 2.8 | .64 | 2.7 | .62 | 2.7 | .60 | 2.6 | .60 |
| 1¼-in. F. W.-Stl. Butt Weld..... | 3.3 | .56 | 3.0 | .52 | 2.8 | .50 | 2.7 | .48 | 2.7 | .46 | 2.6 | .46 |

No valves, headers, stands nor hangers included.
Coils Nos. 5 and 6 are to be made up with couplings and unions.

TABLE 137

BRINE ROOM PIPING. BLACK PIPE

Weight and List Price per Foot of Total Pipe in Coil

| Length of coil —A=..... | 10 Ft. | | 20 Ft. | | 30 Ft. | | 40 Ft. | | 60 Ft. | | 80 Ft. | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Weight, pounds | List, dollars | Weight, pounds | List, dollars | Weight, pounds | List, dollars | Weight, pounds | List, dollars | Weight, pounds | List, dollars | Weight, pounds | List, dollars |
| 2-in. E. H.-W. I. Lap Weld..... | 6.2 | 1.02 | 6.1 | 0.98 | 5.9 | 0.96 | 5.3 | 0.94 | 5.4 | 0.92 | 5.4 | 0.90 |
| 2-in. E. H.-Stl. Butt Weld..... | 6.2 | .64 | 6.1 | .60 | 5.9 | .58 | 5.3 | .56 | 5.4 | .54 | 5.4 | .52 |
| 2-in. F. W.-W. I. Butt Weld.... | 4.8 | .82 | 4.7 | .78 | 4.5 | .76 | 3.9 | .74 | 4.0 | .72 | 4.0 | .70 |
| 2-in. F. W.-Stl. Butt Weld..... | 4.8 | .52 | 4.7 | .48 | 4.5 | .46 | 3.9 | .44 | 4.0 | .42 | 4.0 | .40 |
| 1¼-in. E. H.-W. I. Butt Weld... | 4.0 | .62 | 3.9 | .60 | 3.7 | .58 | 3.2 | .56 | 3.3 | .54 | 3.3 | .52 |
| 1¼-in. E. H.-Stl. Butt Weld..... | 4.0 | .44 | 3.9 | .42 | 3.7 | .40 | 3.2 | .38 | 3.3 | .36 | 3.3 | .34 |
| 1¼-in. F. W.-W. I. Butt Weld... | 3.2 | .52 | 3.1 | .50 | 3.0 | .48 | 2.4 | .46 | 2.5 | .44 | 2.5 | .42 |
| 1¼-in. F. W.-Stl. Butt Weld..... | 3.2 | .36 | 3.1 | .34 | 3.0 | .32 | 2.4 | .30 | 2.5 | .28 | 2.5 | .26 |
| | 1.5 ft. and less | | 2.5 ft. | | 3.5 ft. | | 4.5 ft. | | 5.5 ft. | | 6.5 ft. | |
| 2-in. E. H.-W. I. Lap Weld..... | 6.1 | 1.02 | 6.0 | 1.00 | 5.8 | .96 | 5.2 | .94 | 5.3 | .92 | 5.3 | .90 |
| 2-in. E. H.-Stl. Butt Weld..... | 6.1 | .64 | 6.0 | .62 | 5.8 | .58 | 5.2 | .56 | 5.3 | .54 | 5.3 | .52 |
| 2-in. F. W.-W. I. Lap Weld..... | 4.7 | .80 | 4.6 | .78 | 4.4 | .74 | 3.8 | .72 | 3.9 | .70 | 3.9 | .68 |
| 2-in. F. W.-Stl. Butt Weld..... | 4.7 | .50 | 4.6 | .48 | 4.4 | .44 | 3.8 | .42 | 3.9 | .40 | 3.9 | .38 |
| 1¼-in. E. H.-W. I. Butt Weld... | 3.9 | .62 | 3.8 | .60 | 3.6 | .58 | 3.1 | .56 | 3.2 | .54 | 3.2 | .52 |
| 1¼-in. E. H.-Stl. Butt Weld..... | 3.9 | .42 | 3.8 | .40 | 3.6 | .38 | 3.1 | .36 | 3.2 | .34 | 3.2 | .32 |
| 1¼-in. F. W.-W. I. Butt Weld... | 3.1 | .52 | 3.0 | .50 | 2.9 | .48 | 2.3 | .46 | 2.4 | .44 | 2.4 | .42 |
| 1¼-in. F. W.-Stl. Butt Weld..... | 3.1 | .36 | 3.0 | .34 | 2.9 | .32 | 2.3 | .30 | 2.4 | .28 | 2.4 | .26 |

Weights and prices of coils Nos. 1 and 3 include stands. No valves, headers nor hangers included.

Coils Nos. 1 and 3 are shipped built up with stands or wood strips bolted on. Coils Nos. 2 and 4 are to be made up with couplings and unions.

TABLE 138

DIRECT EXPANSION ROOM PIPING. BLACK PIPE

Weight and List Price per Foot of Total Pipe in Coil

| Length of coil —A= | 10 Ft. | | 20 Ft. | | 30 Ft. | | 40 Ft. | | 60 Ft. | | 80 Ft. | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Weight, pounds | List, dollars | Weight, pounds | List, dollars | Weight, pounds | List, dollars | Weight, pounds | List, dollars | Weight, pounds | List, dollars | Weight, pounds | List, dollars |
| 2-in. E. H.-W. I. Lap Weld | 5.6 | 1.10 | 5.4 | 0.98 | 5.3 | 0.96 | 5.7 | 0.96 | 5.6 | 0.96 | 5.6 | 0.96 |
| 2-in. E. H.-Stl. Butt Weld | 5.6 | .74 | 5.4 | .62 | 5.3 | .60 | 5.7 | .60 | 5.6 | .60 | 5.6 | .60 |
| 2-in. F. W.-W. I. Lap Weld | 4.2 | .90 | 4.0 | .78 | 4.0 | .76 | 4.3 | .76 | 4.2 | .76 | 4.2 | .76 |
| 2-in. F. W.-Stl. Butt Weld | 4.2 | .62 | 4.0 | .50 | 4.0 | .48 | 4.3 | .48 | 4.2 | .48 | 4.2 | .48 |
| 1¼-in. E. H.-W. I. Butt Weld | 3.6 | .66 | 3.4 | .58 | 3.3 | .56 | 3.5 | .56 | 3.5 | .56 | 3.5 | .56 |
| 1¼-in. E. H.-Stl. Butt Weld | 3.6 | .48 | 3.4 | .40 | 3.3 | .38 | 3.5 | .38 | 3.5 | .38 | 3.5 | .38 |
| 1¼-in. F. W.-W. I. Butt Weld | 2.8 | .56 | 2.6 | .48 | 2.5 | .46 | 2.7 | .46 | 2.7 | .46 | 2.7 | .46 |
| 1¼-in. F. W.-Stl. Butt Weld | 2.8 | .42 | 2.6 | .34 | 2.5 | .32 | 2.7 | .32 | 2.7 | .32 | 2.7 | .32 |
| | 1.5 ft. and less | | 2.5 ft. | | 3.5 ft. | | 4.5 ft. | | 5.5 ft. | | 6.5 ft. | |
| 2-in. E. H.-W. I. Lap Weld | 6.0 | 1.50 | 5.7 | 1.26 | 5.6 | 1.18 | 5.5 | 1.16 | 5.4 | 1.14 | 5.4 | 1.14 |
| 2-in. E. H.-Stl. Butt Weld | 6.0 | 1.20 | 5.7 | .98 | 5.6 | .90 | 5.5 | .88 | 5.4 | .86 | 5.4 | .84 |
| 2-in. F. W.-W. I. Lap Weld | ..... | ..... | 4.4 | 1.08 | 4.2 | 1.02 | 4.1 | 1.00 | 4.0 | .98 | 4.0 | .98 |
| 2-in. F. W.-Stl. Butt Weld | ..... | ..... | 4.4 | .84 | 4.2 | .78 | 4.1 | .74 | 4.0 | .72 | 4.0 | .72 |
| 1¼-in. E. H.-W. I. Butt Weld | 4.0 | .76 | 3.6 | .74 | 3.5 | .72 | 3.5 | .70 | 3.4 | .66 | 3.3 | .66 |
| 1¼-in. E. H.-Stl. Butt Weld | 4.0 | .62 | 3.6 | .58 | 3.5 | .56 | 3.5 | .54 | 3.4 | .52 | 3.3 | .52 |
| 1¼-in. F. W.-W. I. Butt Weld | 3.3 | .70 | 3.0 | .66 | 2.8 | .64 | 2.7 | .62 | 2.7 | .60 | 2.6 | .60 |
| 1¼-in. F. W.-Stl. Butt Weld | 3.3 | .58 | 3.0 | .54 | 2.8 | .52 | 2.7 | .50 | 2.7 | .48 | 2.6 | .48 |

No valves, headers, stands nor hangers included. Litharge joints, for soldered joints, see note.
Above prices are for flanges litharged on. For soldering on flanges add per foot as follows:
Coil No. 5, 1¼ in.—.02 list per ft. 2 in.—.03 list per ft.
Coil No. 6, 1¼ in.—.03 list per ft. 2 in.—.04 list per ft.

TABLE 139

DIRECT EXPANSION ROOM PIPING. BLACK PIPE

Weight and List Price per Foot of Total Pipe in Coil.

| Length of coil —A= | 10 Ft. | | 20 Ft. | | 30 Ft. | | 40 Ft. | | 60 Ft. | | 80 Ft. | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Weight, pounds | List, dollars | Weight, pounds | List, dollars | Weight, pounds | List, dollars | Weight, pounds | List, dollars | Weight, pounds | List, dollars | Weight, pounds | List, dollars |
| 2-in. E. H.-W. I. Lap Weld | 6.5 | 1.08 | 6.2 | 1.00 | 6.0 | 0.96 | 5.5 | 0.98 | 5.6 | 0.94 | 5.6 | 0.96 |
| 2-in. E. H.-Stl. Butt Weld | 6.5 | .72 | 6.2 | .64 | 6.0 | .60 | 5.5 | .62 | 5.6 | .58 | 5.6 | .60 |
| 2-in. F. W.-W. I. Lap Weld | 5.2 | .88 | 5.0 | .80 | 4.7 | .76 | 4.2 | .78 | 4.3 | .74 | 4.3 | .76 |
| 2-in. F. W.-Stl. Butt Weld | 5.2 | .60 | 5.0 | .52 | 4.7 | .48 | 4.2 | .50 | 4.3 | .46 | 4.3 | .48 |
| 1¼-in. E. H.-W. I. Butt Weld | 4.2 | .66 | 4.0 | .60 | 3.8 | .58 | 3.2 | .58 | 3.3 | .56 | 3.3 | .56 |
| 1¼-in. E. H.-Stl. Butt Weld | 4.2 | .48 | 4.0 | .42 | 3.8 | .40 | 3.2 | .40 | 3.3 | .38 | 3.3 | .38 |
| 1¼-in. F. W.-W. I. Butt Weld | 3.5 | .56 | 3.3 | .50 | 3.2 | .48 | 2.5 | .48 | 2.6 | .46 | 2.6 | .46 |
| 1¼-in. F. W.-Stl. Butt Weld | 3.5 | .40 | 3.3 | .34 | 3.2 | .32 | 2.5 | .32 | 2.6 | .30 | 2.6 | .30 |
| | 1.5 ft. and less | | 2.5 ft. | | 3.5 ft. | | 4.5 ft. | | 5.5 ft. | | 6.5 ft. | |
| 2-in. E. H.-W. I. Lap Weld | 6.4 | 1.04 | 6.1 | .98 | 5.9 | .96 | 5.4 | .96 | 5.5 | .94 | 5.5 | .96 |
| 2-in. E. H.-Stl. Butt Weld | 6.4 | .68 | 6.1 | .62 | 5.9 | .60 | 5.4 | .60 | 5.5 | .58 | 5.5 | .60 |
| 2-in. F. W.-W. I. Lap Weld | 5.1 | .84 | 4.9 | .78 | 4.6 | .76 | 4.1 | .76 | 4.2 | .74 | 4.2 | .76 |
| 2-in. F. W.-Stl. Butt Weld | 5.1 | .56 | 4.9 | .50 | 4.6 | .48 | 4.1 | .48 | 4.2 | .46 | 4.2 | .48 |
| 1¼-in. E. H.-W. I. Butt Weld | 4.1 | .64 | 3.9 | .58 | 3.7 | .56 | 3.1 | .56 | 3.2 | .54 | 3.2 | .56 |
| 1¼-in. E. H.-Stl. Butt Weld | 4.1 | .46 | 3.9 | .40 | 3.7 | .38 | 3.1 | .38 | 3.2 | .36 | 3.2 | .38 |
| 1¼-in. F. W.-W. I. Butt Weld | 3.4 | .54 | 3.2 | .48 | 3.1 | .46 | 2.4 | .46 | 2.5 | .44 | 2.5 | .46 |
| 1¼-in. F. W.-Stl. Butt Weld | 3.4 | .40 | 3.2 | .34 | 3.1 | .32 | 2.4 | .32 | 2.5 | .30 | 2.5 | .32 |

Weights and prices of coils Nos. 1 and 3 include stands. Weights and prices of coils Nos. 2 and 4 do not include stands. No valves, headers or hangers included. Litharge joints. For soldered joints, see note.

Above prices are for flanges litharged on. For soldering on flanges add per foot as follows:

Coils Nos. 1 and 3, 1¼ in.—.015 list per foot 2 in.—.02 list per foot.

Coils Nos. 2 and 4, 1¼ in.—.025 list per foot 2 in.—.03 list per foot.

TABLE 140

PIPE INSULATION (CORK)

| | Price |
|---|---|
| ½-in. pipe | $1.37 |
| ¾-in. pipe | 1.54 |
| 1 -in. pipe | 1.60 |
| 1¼-in. pipe | 1.65 |
| 1½-in. pipe | 1.68 |
| 2 -in. pipe | 1.88 |
| 2½-in. pipe | 1.99 |
| 3 -in. pipe | 2.17 |

In estimating work of this kind, all fittings to be measured over and counted as an additional foot of pipe.

TABLE 141

STANDARD BELT DRIVE FOR COMPRESSOR

Oak-tanned, Regular Weight, Double-ply Leather

| Size of Tons | Total Length of Belt, Feet | Width, Inches | List Price, Dollars | Selling Price, Dollars |
|---|---|---|---|---|
| 3 | 26 | 5 | 62.50 | 38.00 |
| 4 | 30 | 5 | 72.00 | 44.00 |
| 5 | 30 | 6 | 86.50 | 52.00 |
| 6 | 30 | 6 | 86.50 | 55.00 |
| 8 | 40 | 8 | 154.00 | 95.00 |
| 10 | 40 | 8 | 154.00 | 95.00 |
| 12 | 40 | 10 | 192.00 | 120.00 |
| 15 | 45 | 10 | 215.00 | 130.00 |
| 18 | 45 | 12 | 260.00 | 160.00 |
| 20 | 45 | 12 | 260.00 | 160.00 |
| 25 | 50 | 14 | 288.00 | 175.00 |
| 30 | 50 | 14 | 336.00 | 205.00 |
| 40 | 58 | 14 | 380.00 | 235.00 |
| 50 | 60 | 14 | 400.00 | 245.00 |
| 60 | 60 | 16 | 460.00 | 280.00 |
| 70 | 60 | 16 | 460.00 | 280.00 |
| 80 | 60 | 16 | 460.00 | 280.00 |
| 90 | 80 | 18 | 690.00 | 420.00 |
| 100 | 80 | 18 | 690.00 | 420.00 |
| 110 | 80 | 18 | 690.00 | 420.00 |
| 125 | 100 | 20 | 960.00 | 580.00 |
| 150 | 100 | 22 | 1056.00 | 640.00 |

10-in. belt for 6-in. vertical 28 ft. long, $80.00.

Waterproof belts are recommended for all installations where there is possibility of dampness.

For waterproof belts add 15 per cent to the above list.

TABLE 142

BRINE STRAINERS

| | List Price |
|---|---|
| $1\frac{1}{4}$ in. and $1\frac{1}{2}$ in. | $4 |
| 2 in. and $2\frac{1}{2}$ in. | 15 |
| 3 in. | 22 |
| 4 in. | 30 |

TABLE 143

RAW WATER AIR VALVES, HEADERS AND DROP PIPES

| Size of Ice Can: | List Price |
|---|---|
| 6 ×12 ×29, 50 lb. | $2.70 per can |
| 8 ×16 ×32, 100 lb. | 2.95 per can |
| $11\frac{1}{2}\times22\frac{1}{2}\times32$, 200 lb. | 3.10 per can |
| $11\frac{1}{2}\times22\frac{1}{2}\times45$, 300 lb. | 3.15 per can |
| Hand type core sucker | 19.50 |

Prices include air piping from blower.

TABLE 144

HORIZONTAL BELT-DRIVEN BRINE AGITATORS

Cost Prices

9 in., $75 net, F.O.B. Pennsylvania
12 in., 100 net, F.O.B. Pennsylvania
20 in., 150 net, F.O.B. Pennsylvania.

Each complete with one pulley.

TABLE 145

CALCIUM CHLORIDE

| Solid (650-lb. Barrels) | Per ton | Granulated (375-lb. barrels) | Per ton |
|---|---|---|---|
| 1 ton | $45 | 1 ton | $50 |
| 2 to 3 tons | 42 | 2 to 3 tons | 49 |
| 4 to 8 tons | 40 | 4 to 5 tons | 48 |
| 9 to 15 tons | 38 | 6 to 9 tons | 46 |
| 16 tons and over | 36 | 10 tons and over | 45 |

TABLE 146

PIPING FOR CARBON DIOXIDE

Pipe (Extra Heavy Wrought Steel Pipe) installed, per foot:

| | | | |
|---|---|---|---|
| 3 in | $1.70 | $1\frac{1}{4}$ in | $0.90 |
| $2\frac{1}{2}$ | 1.50 | 1 | .80 |
| 2 | 1.20 | $\frac{3}{4}$ | .70 |
| $1\frac{1}{2}$ | 1.00 | $\frac{1}{2}$ | .60 |

Pipe Covering and Asphaltum Paint Finish, per Foot:

| | | | |
|---|---|---|---|
| 3 in | $4.85 | $1\frac{1}{4}$ in | $2.60 |
| $2\frac{1}{2}$ | 4.20 | 1 | 2.25 |
| 2 | 3.55 | $\frac{3}{4}$ | 1.95 |
| 1 | 3.00 | $\frac{1}{2}$ | 1.75 |

Galvanized Brine Pipe installed, per Foot:

| | | | |
|---|---|---|---|
| 4 in | $1.83 | $1\frac{1}{2}$ in | $0.98 |
| 3 | 1.68 | $1\frac{1}{4}$ | .88 |
| $2\frac{1}{2}$ | 1.43 | 1 | .78 |
| 2 | 1.18 | $\frac{3}{4}$ | .68 |
| | | $\frac{1}{2}$ | .58 |

Galvanized Brine Pipe, Asphaltum Paint and Covering, per Foot:

| | | | |
|---|---|---|---|
| 4 in | $5.98 | $1\frac{1}{2}$ in | $2.58 |
| 3 | 4.83 | $1\frac{1}{4}$ | 2.23 |
| $2\frac{1}{2}$ | 4.18 | 1 | 1.93 |
| 2 | 3.53 | $\frac{3}{4}$ | 1.73 |

TABLE 147

"National" Wrought Steel Pipe

Discounts F.O.B. Chicago

| Size | Random Pipe | | Size | Random Pipe | |
|---|---|---|---|---|---|
| | Black | Galvanized | | Black | Galvanized |
| ⅛ in., butt | 37 | 4 | 2 in., lap | 41 | 28 |
| ¼ and ⅜ in. butt | 43 | 16 | 2½ to 6 in., lap | 51 | 38 |
| ½ in., butt | 48 | 33 | 7 to 8 in., lap | 43 | 29 |
| ¾ in., butt | 52 | 39 | 9 and 10 in., lap | 41 | 27 |
| 1 to 3 in., butt | 54 | 41 | 11 and 12 in., lap. | 40 | 26 |
| Extra Strong Plain Ends | | | | | |
| ⅛ in., butt | 25 | | 2 in., lap | 37 | 13 |
| ¼ and ⅜ in., butt. | 31 | | 2½ to 4 in., lap | 41 | 17 |
| ½ in., butt | 37 | 13 | 4½ to 6 in., lap | 40 | 16 |
| ¾ in., butt | 42 | 18 | 7 to 8 in., lap | 34 | 10 |
| 1 to 1½ in., butt | 44 | 20 | 9 and 10 in., lap. | On application | |
| 2 to 3 in., butt | 45 | 21 | 11 and 12 in., lap | | |
| Double Extra Strong Plain Ends | | | | | |
| ½ in., butt | 22 | On Application | 2½ to 4 in., lap | 23 | On Application |
| ¾ to 1½ in., butt. | 25 | | 4½ to 6 in. lap | 22 | |
| 2 and 2½ in. butt | 27 | | 7 and 8 in. lap | 16 | |
| 2 in. lap | 21 | | | | |
| Net Price per 100 Ft. Standard Weight F.O.B. Chicago | | | | | |
| ⅛ in. butt | $3.47 | $5.28 | 4 in. lap | $53.41 | $67.58 |
| ¼ in. butt | 3.4 | 5.04 | 4½ in. lap | 62.23 | 78.74 |
| ⅜ in. butt | 3.42 | 5.04 | 5 in. lap | 72.52 | 91.76 |
| ½ in. butt | 4.42 | 5.70 | 6 in. lap | 94.08 | 119.04 |
| ¾ in. butt | 5.52 | 7.02 | 7 in. lap | 135.66 | 168.98 |
| 1 in. butt | 7.82 | 10.03 | 8 in. 25 lb. lap. | 142.50 | 177.50 |
| 1¼ in. butt | 10.58 | 13.57 | 8 in. 28 lb., lap. | 164.16 | 204.48 |
| 1½ in., butt | 12.65 | 16.23 | 9 in., lap | 203.55 | 251.85 |
| 2 in., butt | 17.02 | 21.83 | 10 in., 35 lb., lap. | 206.50 | 255.50 |
| 2½ in., butt | 26.91 | 34.52 | 10 in., 41 lb., lap. | 243.08 | 300.76 |
| 3 in., butt | 35.19 | 45.14 | 12 in., 45 lb., lap. | 270.00 | 333.00 |
| 2 in., lap | 21.83 | 26.64 | 12 in., 50 lb., lap. | 304.20 | 375.18 |
| 3½ in., lap | 45.08 | 57.04 | | | |

Weight of 8 in.–10 in.–12 in. must be specified.

Tarred pipe, all sizes, and O.D. pipe 14 to 24 in. in stock.
Prices on application.

TABLE 147*a*

"Reading" Wrought Iron Pipe

Discounts F.O.B. Chicago

| Size | Random Pipe | | Size | Random Pipe | |
|---|---|---|---|---|---|
| | Black | Galvanized | | Black | Galvanized |
| ⅛ in. | On application | | 2 in., lap | 7 | +11 |
| ¼ and ⅜ in., butt. | +27 | +57 | 2½ in., lap | 10 | + 7 |
| ½ in., butt | 6 | +16 | 3 to 6 in., lap | 12 | + 5 |
| ¾ in., butt | 12 | + 7 | 7 to 12 in., lap | 8 | + 9 |
| 1 to 1½ in., butt. | 14 | + 5 | | | |

Extra Strong Plain Ends

| | | | | | |
|---|---|---|---|---|---|
| ¼ and ⅜ in., butt. | +40 | On Application | 2½ in., lap | 8 | On Application |
| ½ in., butt | Net list | | 3 to 4 in., lap | 8 | |
| ¾ in., butt | 7 | | 4½ to 6 in., lap | 7 | |
| 1 to 1½ in., butt. | 9 | | 7 and 8 in., lap | + 2 | |
| 2 in., lap | 2 | | 9 to 12 in., lap | + 7 | |

Double Extra Strong Plain Ends

| | | | |
|---|---|---|---|
| ½ in., butt<br>¾ to 1½ in., butt.<br>2 in., lap | On application | 2½ to 4 in., lap<br>4½ to 6 in., lap | On application |

Net Price per 100 Ft., Standard Weight, F.O.B. Chicago

| | | | | | |
|---|---|---|---|---|---|
| ¼ in., butt | $7.62 | $9.42 | 4 in., lap | $95.92 | $114.45 |
| ⅜ in., butt | 7.62 | 9.42 | 4½ in., lap | 111.76 | 133.35 |
| ½ in., butt | 7.99 | 9.86 | 5 in., lap | 130.24 | 155.40 |
| ¾ in., butt | 10.12 | 12.31 | 6 in., lap | 168.96 | 201.60 |
| 1 in., butt | 14.62 | 17.85 | 7 in., lap | 218.90 | 259.42 |
| 1¼ in., butt | 19.78 | 24.15 | 8 in., 25 lb., lap. | | |
| 1½ in., butt | 23.65 | 28.88 | 8 in., 28 lb., lap. | 264.96 | 313.92 |
| 1½ in., lap | | | 9 in., lap | 317.40 | 367.05 |
| 2 in., lap | 34.41 | 41.07 | 10 in., 35 lb., lap. | | |
| 2½ in., lap | 52.65 | 62.60 | 10 in., 41 lb., lap. | 379.04 | 449.08 |
| 3 in., lap | 67.32 | 80.33 | 12 in., 45 lb., lap. | | |
| 3½ in., lap | 80.96 | 96.60 | 12 in., 50 lb., lap. | 466.44 | 552.63 |

Reading wrought iron O.D. pipe up to 20 in. can be furnished on short notice.

TABLE 147*b*

"Alpha" Brass Pipe and Tubing

(Seamless)

Standard Iron Pipe Size—Net Prices per Foot

| Size, Inches | Full Lengths | Cut Lengths | Add for Nickel Plating | Threads Each |
|---|---|---|---|---|
| $\frac{1}{8}$ | .11 | .14 | .08 | .07$\frac{1}{2}$ |
| $\frac{1}{4}$ | .18 | .23 | .08 | .07$\frac{1}{2}$ |
| $\frac{3}{8}$ | .23 | .29 | .08 | .07$\frac{1}{2}$ |
| $\frac{1}{2}$ | .31 | .34 | .13 | .07$\frac{1}{2}$ |
| $\frac{3}{4}$ | .39 | .49 | .16 | .07$\frac{1}{2}$ |
| 1 | .55 | .69 | .19 | .09 |
| 1$\frac{1}{4}$ | .80 | 1.00 | .20 | .10$\frac{1}{2}$ |
| 1$\frac{1}{2}$ | .95 | 1.18 | .21 | .12 |
| 2 | 1.25 | 1.57 | .28 | .15 |
| 2$\frac{1}{2}$ | 1.82 | 2.73 | .39 | .27$\frac{1}{2}$ |
| 3 | 2.60 | 3.90 | .47 | .30 |

Larger sizes on application.

| Size O.D., Inches | Seamless Tubing No. 17G Nickel Plated, 12 Ft. Lengths, Net per Foot | Size Pipe to Cover, Inches | Tubing (Casing) Nickel Plated to Cover Wrought Pipe, Net per Foot |
|---|---|---|---|
| 1$\frac{1}{4}$ | .46 | $\frac{3}{8}$ | .25 |
| 1$\frac{3}{8}$ | .50 | $\frac{1}{2}$ | .30 |
| 1$\frac{1}{2}$ | .60 | $\frac{3}{4}$ | .35 |
| 1$\frac{3}{4}$ | ..... | 1 | .45 |
| 2 | ..... | 1$\frac{1}{4}$ | .55 |
| | No. 20G Rough in 12 Ft. Lengths | 1$\frac{1}{2}$ | .65 |
| 1$\frac{1}{2}$ | .35 | | For cutting add 20 per cent to above. |

Threading, all sizes of tubing, .25 net per thread.

TABLE 148

Ice Can Covers Only

| Quantity | Dimensions | Price | Remarks |
|---|---|---|---|
| 50 | $8\frac{1}{4}\times14\frac{1}{4}$ | $0.75 | 3–$\frac{7}{8}$ in. lumber water proof paper between. |
| 100 | $10\frac{1}{4}\times18\frac{1}{4}$ | 1.25 | |
| 200 | $13\frac{5}{8}\times24\frac{5}{8}$ | 2.50 | |
| 300 | $13\frac{5}{8}\times24\frac{5}{8}$ | 2.50 | |

TABLE 149

Automatic Sprinkler

| Size of Cans, Pounds | Automatic Sprinkling Pumps and Pans Complete | | |
|---|---|---|---|
| | Floor space | Pan depth, inches | Price |
| 50 | 2 ft. 0 in.×3 ft. 9 in. | 4 | $70 |
| 100 | 2 ft. 8 in.×4 ft. 10 in. | 5 | 75 |
| 200 | 3 ft. 0 in.×5 ft. 2½ in. | 5 | 95 |
| 300 | 3 ft. 0 in.×6 ft. 0 in. | 5 | 110 |

TABLE 150

Synchronous Motors

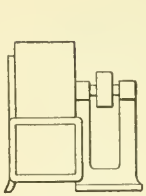
Including motor with base and 2 pedestals for coupling to compressors; also push button across the line, auto-starter and including motor generator exciter sets.

75 hp. 100 per cent P.F. 200 r.p.m., 2 pedestal bearings base and shaft with extension for coupling to compressor across line starter from push button, indicating a.c. ammeter and d.c. ammeter..... $3150
150 hp., 200 r.p.m., same as above..... 3800
225 hp., 200 r.p.m., same as above..... 4250
150 hp., 180 r.p.m., same as above..... 4160
300 hp., 171 r.p.m., same as above..... 5480
450 hp., 164 r.p.m., same as above..... 6160

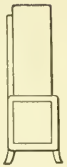
Without shafts or bearings, but including pole plates, split rotor exciter sets and automatic control push button starter:

150 hp., 100 per cent, P.F., 150 r.p.m., 220 volts..... $4435
150 hp., 100 per cent, P.F., 138 r.p.m., 220 volts..... 4825
200 hp., 100 per cent, P.F., 150 r.p.m., 220 volts..... 4640
200 hp., 100 per cent, P.F., 138 r.p.m., 220 volts..... 4960

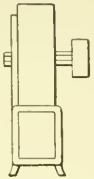
TABLE 151



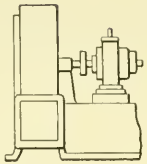
Housing, Pedestal, 2-Bearings, pulley Shaft, Wheel
(1)



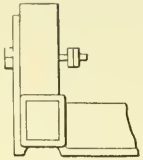
Housing & Wheel only
(2)



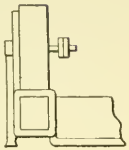
Housing, Wheel & Pedestal for Motor or Engine
(3)



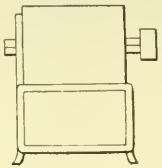
Housing, Wheel & Pedestal for Motor or Engine
(4)



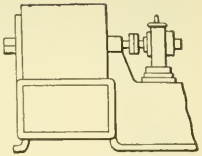
Housing, Wheel Shaft One Intermediate Bearing, Coupling & Pedestal
(5)



Housing, Wheel Shaft, One Bearing (in inlet) Pedestal & Coupling
(6)



Double Inlet
(3)



Double Inlet
(7)

| Fan Number | Diameter of Wheel, Inches | Arrangement No. 1 | | Single Inlet | | | | | | | | Double Inlet | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Arrangement No. 2 | | Arrangement No. 3 | | Arrangement No. 4 | | Arrangement Nos. 5 and 6 | | Arrangement No. 3 | | Arrangement No. 7 | |
| | | Price, dollars | Weight, pounds | Price, dollars | Weight, pounds | Price, dollars | Weight, pounds | Price, dollars | Weight, pounds | Price, dollars | Weight, pounds | Price, dollars | Weight, pounds | Price, dollars | Weight, pounds |
| 00 | 3 | 34 | 10 | 32 | 8 | ..... | ..... | 34 | 10 | | | | | | |
| 0 | 4½ | 50 | 30 | 48 | 25 | ..... | ..... | 50 | 30 | | | | | | |
| 1 | 6 | 76 | 60 | 70 | 50 | ..... | ..... | 76 | 60 | | | | | | |
| 1¼ | 7½ | 102 | 90 | 96 | 70 | ..... | ..... | 102 | 80 | | | | | | |
| 1½ | 9 | 134 | 130 | 128 | 100 | ..... | ..... | 134 | 120 | | | | | | |
| 2 | 12 | 168 | 200 | 160 | 170 | ..... | ..... | 168 | 225 | | | | | | |
| 2½ | 15 | 210 | 320 | 200 | 280 | ..... | ..... | 210 | 370 | ..... | ..... | 354 | 520 | 378 | 550 |
| 3 | 18 | 270 | 500 | 256 | 400 | ..... | ..... | 270 | 600 | ..... | ..... | 454 | 625 | 488 | 650 |
| 3½ | 21 | 354 | 700 | 336 | 500 | ..... | ..... | 354 | 730 | ..... | ..... | 588 | 775 | 630 | 775 |
| 4 | 24 | 482 | 850 | 414 | 650 | 440 | 620 | 460 | 620 | 482 | 700 | 722 | 1050 | 772 | 1130 |
| 4½ | 27 | 556 | 1020 | 480 | 750 | 500 | 800 | 530 | 800 | 556 | 900 | 840 | 1375 | 890 | 1500 |
| 5 | 30 | 628 | 1350 | 542 | 950 | 570 | 1050 | 600 | 1050 | 628 | 1200 | 950 | 1950 | 1010 | 2150 |
| 6 | 36 | 776 | 1800 | 672 | 1200 | 706 | 1500 | 740 | 1500 | 776 | 1700 | 1180 | 3000 | 1240 | 3200 |

Sirocco fans:
Arrangements Nos. 1, 2 and 3
Arrangements Nos. 4, 5, 6 and 7
Add 5 per cent for Troy type. Size 4 and larger.

Discount to users
54 per cent
51 per cent

Discount to dealers
60 per cent
58 per cent

# TABLE 152

## Type CS Squirrel-Cage Motors

### Open Type, 60 Cycles, Continuous Rated 40° C., Constant Speed

| Horse-power | Volts | Number of Poles | Full Load, R.p.m. | Frame | Price, Dollars: Complete motor with: Type "A" manual starter Class 10,700 | Price, Dollars: Complete motor with: Magnetic auto-transformer type starter | Price, Dollars: Bare motor with: Type "A" manual starter Class 10,700 | Price, Dollars: Bare motor with: Magnetic auto-transformer type starter | Price, Dollars: Bare motor | Price, Dollars: Omission or addition for: Rails | Price, Dollars: Omission or addition for: Pulley | Price, Dollars: Addition for vertical parts | Approximate Shipping Weight, Pounds: Complete motor with: Type A starter | Approximate Shipping Weight, Pounds: Complete motor with: Magnetic starter | Approximate Shipping Weight, Pounds: Bare motor | Approximate Shipping Weight, Pounds: Omission or addition for: Rails | Approximate Shipping Weight, Pounds: Omission or addition for: Pulley | Pulley Dimensions, Inches: Standard diameter and face | Pulley Dimensions, Inches: Bore | Pulley Dimensions, Inches: Special pulley, minimum diameter and maximum face | Shaft Dimensions, Inches: Standard shaft extension | Shaft Dimensions, Inches: Standard keyway dimensions, L. W. D. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 7½ | 220 | 2 | 3450 | 243C | ..... | ..... | 193 | 257 | 117 | 6 | ..... | 60 | 422*c* | 422*c* | 192 | 30 | ..... | ..... | 1¼ | ..... | 3 11/16 | 2 15/16 × 15/16 × 1/4 |
| | | 4 | 1750 | 250C | 190 | 254 | 182 | 246 | 106 | 6 | 2 | 60 | 339 | 439 | 199 | 35 | 5 | 5×4½ | 1¼ | 5×5 | 3 13/16 | 2 15/16 × 15/16 × 3/8 |
| | | 6 | 1160 | 352C | 224 | 288 | 210 | 274 | 134 | 11 | 3 | 75 | 443 | 543 | 291 | 41 | 10 | 6×5 | 1⅝ | 4½×9 | 5 3/4 | 4½ × 3/8 × 1/4 |
| | 440 | 8 | 870 | 370C | 259 | 323 | 245 | 309 | 169 | 11 | 3 | 75 | 510 | 610 | 348 | 47 | 15 | 7×6 | 1⅝ | 5×10 | 5 1/16 | 4½ × 3/8 × 1/4 |
| | | 10 | 690 | 460C | 313 | 377 | 298 | 362 | 222 | 11 | 4 | 80 | 588 | 688 | 425 | 48 | 15 | 8×7 | 1⅞ | 6×10 | 6 1/16 | 5/8 × 7/16 × 5/32 |
| | | 12 | 575 | 464C | 334 | 398 | 319 | 383 | 243 | 11 | 4 | 80 | 625 | 725 | 462 | 48 | 15 | 8×7 | 1¼ | 6×10 | 6 1/16 | 5/8 × 7/16 × 5/32 |
| | 550 | 14 | 490 | 566C | 380 | 444 | 364 | 428 | 288 | 12 | 4 | 110 | 940 | 1040 | 735 | 80 | 25 | 9×8 | 2⅜ | 7×12 | 8 | 7/8 × 9/16 × 3/16 |
| | | 16 | 435 | 586C | 392 | 456 | 376 | 440 | 300 | 12 | 4 | 110 | 1041 | 1141 | 836 | 80 | 25 | 9×8 | 2⅜ | 9×12 | 8 | 7/8 × 9/16 × 3/16 |
| 10 | 220 | 2 | 3450 | 351.5C | ..... | ..... | 215 | 275 | 135 | 11 | ..... | 75 | 398*c* | 498*c* | 256 | 42 | ..... | ..... | 1⅝ | ..... | 4¼ | 4½ × 3/8 × 1/8 |
| | | 4 | 1750 | 351C | 217 | 277 | 203 | 263 | 123 | 11 | 3 | 75 | 408 | 508 | 256 | 42 | 10 | 6×5 | 1⅝ | 4×9 | 5 3/4 | 4½ × 3/8 × 1/4 |
| | | 6 | 1160 | 370C | 251 | 311 | 237 | 297 | 157 | 11 | 3 | 75 | 510 | 610 | 348 | 47 | 15 | 7×6 | 1⅝ | 5×10 | 5 1/16 | 4½ × 3/8 × 1/4 |
| | 440 | 8 | 870 | 460C | 287 | 347 | 272 | 332 | 192 | 11 | 4 | 80 | 588 | 688 | 425 | 48 | 15 | 8×7 | 1⅞ | 6×10 | 6 1/16 | 5/8 × 7/16 × 5/32 |
| | | 10 | 690 | 480C | 350 | 410 | 335 | 395 | 255 | 11 | 4 | 80 | 699 | 799 | 536 | 48 | 15 | 8×7 | 1⅞ | 7×10 | 6 1/16 | 5/8 × 7/16 × 5/32 |
| | | 12 | 575 | 566C | 376 | 436 | 360 | 420 | 280 | 12 | 4 | 110 | 940 | 1040 | 735 | 80 | 25 | 9×8 | 2⅜ | 7×12 | 8 | 7/8 × 9/16 × 3/16 |
| | 550 | 14 | 490 | 586C | 417 | 477 | 400 | 460 | 320 | 12 | 5 | 110 | 1056 | 1156 | 836 | 80 | 40 | 10×9 | 2¾ | 9×12 | 8 | 7/8 × 9/16 × 3/16 |
| | | 16 | 435 | 642C | 446 | 506 | 425 | 485 | 345 | 16 | 5 | 115 | 1607 | 1705 | 1302 | 165 | 40 | 10×9 | 2⅞ | 9×14 | 9½ | 8¼ × 11/16 × 1/4 |
| 15 | 220 | 2 | 3450 | 363.5C | ..... | ..... | 251 | 309 | 169 | 11 | ..... | 75 | 560*c* | 610*c* | 363 | 47 | ..... | ..... | 1⅝ | ..... | 4½ | 4½ × 3/8 × 1/8 |
| | | 4 | 1750 | 371C | 252 | 310 | 238 | 296 | 156 | 11 | 3 | 75 | 570 | 620 | 358 | 47 | 15 | 7×6 | 1⅝ | 5×10 | 5 1/16 | 4½ × 3/8 × 1/4 |
| | | 6 | 1160 | 460C | 294 | 352 | 279 | 337 | 197 | 11 | 4 | 80 | 638 | 688 | 425 | 48 | 15 | 8×7 | 1⅞ | 6×10 | 6 1/16 | 5/8 × 7/16 × 5/32 |
| | 440 | 8 | 870 | 480C | 330 | 388 | 315 | 373 | 233 | 11 | 4 | 80 | 759 | 809 | 536 | 48 | 25 | 9×8 | 1⅞ | 6×10 | 6 1/16 | 5/8 × 7/16 × 5/32 |
| | | 10 | 690 | 566C | 410 | 468 | 393 | 451 | 311 | 12 | 5 | 110 | 1005 | 1055 | 735 | 80 | 40 | 10×9 | 2⅜ | 8×12 | 8 | 7/8 × 9/16 × 3/16 |
| | 550 | 12 | 575 | 582C | 440 | 498 | 423 | 481 | 341 | 12 | 5 | 110 | 1074 | 1124 | 804 | 80 | 40 | 10×9 | 2¾ | 9×12 | 8 | 7/8 × 9/16 × 3/16 |
| | | 14 | 490 | 642C | 495 | 553 | 472 | 530 | 390 | 16 | 7 | 115 | 1667 | 1717 | 1302 | 165 | 50 | 11×10 | 2⅞ | 9×14 | 9½ | 8¼ × 11/16 × 1/4 |
| | | 16 | 435 | 644C | 525 | 583 | 502 | 560 | 420 | 16 | 7 | 115 | 1744 | 1794 | 1379 | 165 | 50 | 11×10 | 2⅞ | 9×14 | 9½ | 8¼ × 11/16 × 1/4 |

| | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 2 | 3450 | 461.5C | ..... | ..... | 288 | 354 | 204 | 11 | ..... | 80 | 662*c* | 712*c* | 464 | 48 | ..... | ..... | $1\frac{7}{8}$ | ..... | $4\frac{1}{2}$ | $5\frac{5}{8}\times\frac{7}{16}\times\frac{5}{32}$ |
| | | 4 | 1750 | 461C | 287 | 353 | 272 | 338 | 188 | 11 | 4 | 80 | 677 | 727 | 464 | 48 | 15 | 8×7 | $1\frac{7}{8}$ | $5\frac{1}{2}\times10$ | $6\frac{1}{16}$ | $5\frac{5}{8}\times\frac{7}{16}\times\frac{5}{32}$ |
| | | 6 | 1160 | 480C | 331 | 397 | 316 | 382 | 232 | 11 | 4 | 80 | 759 | 809 | 536 | 48 | 25 | 9×8 | $1\frac{7}{8}$ | 7×10 | 8 | $6\frac{5}{8}\times\frac{9}{16}\times\frac{3}{16}$ |
| | 220 | 8 | 870 | 566C | 372 | 438 | 355 | 421 | 271 | 12 | 5 | 110 | 1005 | 1055 | 735 | 80 | 40 | 10×9 | $2\frac{3}{8}$ | 7×12 | 8 | $6\frac{5}{8}\times\frac{9}{16}\times\frac{3}{16}$ |
| | 440 | 10 | 690 | 586C | 462 | 528 | 443 | 509 | 359 | 12 | 7 | 110 | 1116 | 1166 | 836 | 80 | 50 | 11×10 | $2\frac{3}{8}$ | 9×12 | 8 | $6\frac{5}{8}\times\frac{9}{16}\times\frac{3}{16}$ |
| | 550 | 12 | 575 | 642C | 501 | 567 | 478 | 544 | 394 | 16 | 7 | 115 | 1667 | 1717 | 1302 | 165 | 50 | 11×10 | $2\frac{7}{8}$ | 9×14 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| 20 | | 14 | 490 | 644C | 561 | 627 | 534 | 600 | 450 | 16 | 11 | 115 | 1759 | 1809 | 1369 | 165 | 75 | 12×12 | $2\frac{7}{8}$ | 9×14 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 16 | 435 | 750C | 611 | 677 | 579 | 645 | 495 | 21 | 11 | 120 | 2480 | 2550 | 2090 | 165 | 75 | 12×12 | $3\frac{3}{8}$ | 12×14 | $10\frac{1}{4}$ | $8\frac{5}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 4 | 1750 | 565C | 461 | 861 | 445 | 845 | 295 | 12 | 4 | 110 | 1135 | 2335 | 740 | 80 | 15 | 8×7 | $1\frac{7}{8}$ | $5\frac{1}{2}\times10$ | 8 | $5\frac{5}{8}\times\frac{7}{16}\times\frac{9}{32}$ |
| | 2200 | 6 | 1160 | 567C | 516 | 916 | 500 | 900 | 350 | 12 | 4 | 110 | 1168 | 2368 | 763 | 80 | 25 | 9×8 | $1\frac{7}{8}$ | 7×10 | 8 | $6\frac{5}{8}\times\frac{9}{16}\times\frac{9}{32}$ |
| | | 8 | 870 | 642C | 565 | 965 | 544 | 944 | 394 | 16 | 5 | 115 | 1807 | 3007 | 1302 | 165 | 40 | 10×9 | $2\frac{3}{8}$ | 7×12 | $9\frac{1}{2}$ | $6\frac{5}{8}\times\frac{9}{16}\times\frac{3}{16}$ |
| | | 2 | 3450 | 473.5C | ..... | ..... | 324 | 388 | 238 | 11 | ..... | 80 | 711*c* | 761*c* | 513 | 48 | ..... | ..... | $1\frac{7}{8}$ | ..... | $4\frac{1}{2}$ | $5\frac{5}{8}\times\frac{7}{16}\times\frac{5}{32}$ |
| | | 4 | 1750 | 473C | 318 | 382 | 303 | 367 | 217 | 11 | 4 | 80 | 718 | 768 | 495 | 48 | 25 | 9×8 | $1\frac{7}{8}$ | 6×10 | $6\frac{11}{16}$ | $5\frac{5}{8}\times\frac{7}{16}\times\frac{5}{32}$ |
| | | 6 | 1160 | 567C | 369 | 433 | 352 | 416 | 266 | 12 | 5 | 110 | 1033 | 1083 | 763 | 80 | 40 | 10×9 | $2\frac{3}{8}$ | 7×12 | 8 | $6\frac{5}{8}\times\frac{9}{16}\times\frac{3}{16}$ |
| | 220 | 8 | 870 | 586C | 415 | 479 | 396 | 460 | 310 | 12 | 7 | 110 | 1116 | 1166 | 836 | 80 | 50 | 11×10 | $2\frac{3}{8}$ | 8×12 | 8 | $6\frac{5}{8}\times\frac{9}{16}\times\frac{3}{16}$ |
| | 440 | 10 | 690 | 644C | 511 | 575 | 488 | 552 | 402 | 16 | 7 | 115 | 1734 | 1784 | 1369 | 165 | 50 | 11×10 | $2\frac{7}{8}$ | 9×14 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | 550 | 12 | 575 | 750C | 562 | 626 | 530 | 594 | 444 | 21 | 11 | 120 | 2480 | 2530 | 2090 | 165 | 75 | 12×12 | $3\frac{3}{8}$ | 12×14 | $10\frac{1}{4}$ | $8\frac{5}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| 25 | | 14 | 490 | 750C | 628 | 692 | 593 | 657 | 507 | 21 | 14 | 120 | 2500 | 2550 | 2090 | 165 | 95 | 14×12 | $3\frac{3}{8}$ | 12×14 | $10\frac{1}{4}$ | $8\frac{5}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 16 | 435 | 752C | 678 | 742 | 643 | 707 | 557 | 21 | 14 | 120 | 2527 | 2577 | 2117 | 165 | 95 | 14×12 | $3\frac{3}{8}$ | 12×14 | $10\frac{1}{4}$ | $8\frac{5}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 4 | 1750 | 565C | 501 | 901 | 485 | 885 | 330 | 12 | 4 | 110 | 1145 | 2345 | 740 | 80 | 25 | 9×8 | $2\frac{3}{8}$ | 6×10 | 8 | $6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$ |
| | 2200 | 6 | 1160 | 642C | 552 | 952 | 531 | 931 | 376 | 16 | 5 | 115 | 1807 | 3007 | 1302 | 165 | 40 | 10×9 | $2\frac{7}{8}$ | 7×12 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 8 | 870 | 644C | 610 | 1010 | 587 | 987 | 432 | 16 | 7 | 115 | 1884 | 3084 | 1369 | 165 | 50 | 11×10 | $2\frac{7}{8}$ | 9×12 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 2 | 3450 | 473.5C | ..... | ..... | 368 | 430 | 280 | 11 | ..... | 80 | 711*c* | 761*c* | 713 | 48 | ..... | ..... | $1\frac{7}{8}$ | ..... | $4\frac{1}{2}$ | $5\frac{5}{8}\times\frac{7}{16}\times\frac{5}{32}$ |
| | | 4 | 1750 | 481C | 348 | 410 | 333 | 395 | 245 | 11 | 4 | 80 | 751 | 801 | 528 | 48 | 25 | 9×8 | $1\frac{7}{8}$ | 7×10 | $6\frac{11}{16}$ | $5\frac{5}{8}\times\frac{7}{16}\times\frac{5}{32}$ |
| | | 6 | 1160 | 567C | 404 | 466 | 385 | 447 | 297 | 12 | 7 | 110 | 1053 | 1103 | 763 | 80 | 50 | 11×10 | $2\frac{3}{8}$ | 8×12 | 8 | $6\frac{5}{8}\times\frac{9}{16}\times\frac{3}{16}$ |
| | 220 | 8 | 870 | 642C | 458 | 520 | 435 | 497 | 347 | 16 | 7 | 115 | 1667 | 1717 | 1302 | 165 | 50 | 11×10 | $2\frac{7}{8}$ | 9×12 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 10 | 690 | 644C] | 559 | 621 | 532 | 594 | 444 | 16 | 11 | 115 | 1759 | 1809 | 1369 | 165 | 75 | 12×12 | $2\frac{7}{8}$ | 10×14 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 12 | 575 | 750C | 614 | 676 | 579 | 641 | 491 | 21 | 14 | 120 | 2500 | 2550 | 2090 | 165 | 95 | 14×12 | $3\frac{3}{8}$ | 12×14 | $10\frac{1}{4}$ | $8\frac{5}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 14 | 490 | 752C | 683 | 745 | 648 | 710 | 560 | 21 | 14 | 120 | 2527 | 2577 | 2117 | 165 | 95 | 14×12 | $3\frac{3}{8}$ | 12×14 | $10\frac{1}{4}$ | $8\frac{5}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 16 | 435 | 752C | 744 | 806 | 704 | 766 | 616 | 21 | 19 | 120 | 2555 | 2605 | 2115 | 165 | 125 | 16×13 | $3\frac{3}{8}$ | 12×14 | $10\frac{1}{4}$ | $8\frac{5}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 2 | 3450 | 473.5C | ..... | ..... | 368 | 430 | 280 | 11 | ..... | 80 | 711*c* | 761*c* | 513 | 48 | ..... | ..... | $1\frac{7}{8}$ | ..... | $4\frac{1}{2}$ | $5\frac{5}{8}\times\frac{7}{16}\times\frac{5}{32}$ |
| | | 4 | 1750 | 481C | 348 | 410 | 333 | 395 | 245 | 11 | 4 | 80 | 751 | 801 | 528 | 48 | 25 | 9×8 | $1\frac{7}{8}$ | 7×10 | $6\frac{11}{16}$ | $5\frac{5}{8}\times\frac{7}{16}\times\frac{5}{32}$ |
| | | 6 | 1160 | 567C | 404 | 466 | 385 | 447 | 297 | 12 | 7 | 110 | 1053 | 1103 | 763 | 80 | 50 | 11×10 | $2\frac{3}{8}$ | 8×12 | 8 | $6\frac{5}{8}\times\frac{9}{16}\times\frac{3}{16}$ |
| | 440 | 8 | 870 | 642C | 458 | 520 | 435 | 497 | 347 | 16 | 7 | 115 | 1667 | 1717 | 1302 | 165 | 50 | 11×10 | $2\frac{7}{8}$ | 9×12 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | 550 | 10 | 690 | 644C | 559 | 621 | 532 | 594 | 444 | 16 | 11 | 115 | 1759 | 1809 | 1369 | 165 | 75 | 12×12 | $2\frac{7}{8}$ | 10×14 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| 30 | | 12 | 575 | 750C] | 614 | 676 | 579 | 641 | 491 | 21 | 14 | 120 | 2500 | 2550 | 2090 | 165 | 95 | 14×12 | $3\frac{3}{8}$ | 12×14 | $10\frac{1}{4}$ | $8\frac{5}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 14 | 490 | 752C | 683 | 745 | 648 | 710 | 560 | 21 | 14 | 120 | 2527 | 2577 | 2117 | 165 | 95 | 14×12 | $3\frac{3}{8}$ | 12×14 | $10\frac{1}{4}$ | $8\frac{5}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 16 | 435 | 752C | 744 | 806 | 704 | 766 | 616 | 21 | 19 | 120 | 2555 | 2605 | 2115 | 165 | 125 | 16×13 | $3\frac{3}{8}$ | 12×14 | $10\frac{1}{4}$ | $8\frac{5}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 4 | 1750 | 565C | 515 | 910 | 499 | 894 | 339 | 12 | 4 | 110 | 1145 | 2345 | 740 | 80 | 25 | 9×8 | $2\frac{3}{8}$ | 6×12 | 8 | $6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$ |
| | 2200 | 6 | 1160 | 642C | 595 | 990 | 572 | 967 | 412 | 16 | 7 | 115 | 1817 | 3017 | 1302 | 165 | 50 | 11×10 | $2\frac{7}{8}$ | 8×12 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 8 | 870 | 644C | 650 | 1045 | 627 | 1022 | 467 | 16 | 7 | 115 | 1884 | 3084 | 1369 | 165 | 50 | 11×10 | $2\frac{7}{8}$ | 9×12 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 10 | 690 | 658C | 770 | 1165 | 743 | 1138 | 583 | 16 | 11 | 115 | 2156 | 3456 | 1716 | 165 | 75 | 12×12 | $2\frac{7}{8}$ | 11×14 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |

For notes and explanation of reference letters, see last page of this table.

TABLE 152—*Continued*

| Horse-power | Volts | Number of Poles | Full Load, R.p.m. | Frame | Price, Dollars: Complete motor with: Type "A" manual starter Class 10,700 | Price, Dollars: Complete motor with: Magnetic auto-transformer type starter | Price, Dollars: Bare motor with: Type "A" manual starter Class 10,700 | Price, Dollars: Bare motor with: Magnetic auto-transformer type starter | Price, Dollars: Bare motor | Price, Dollars: Omission or addition for: Rails | Price, Dollars: Omission or addition for: Pulley | Price, Dollars: Addition for vertical parts | Approximate Shipping Weight, Pounds: Complete motor with: Type A starter | Approximate Shipping Weight, Pounds: Complete motor with: Magnetic starter | Approximate Shipping Weight, Pounds: Bare motor | Approximate Shipping Weight, Pounds: Omission or addition for: Rails | Approximate Shipping Weight, Pounds: Omission or addition for: Pulley | Pulley Dimensions, Inches: Standard diameter and face | Pulley Dimensions, Inches: Bore | Pulley Dimensions, Inches: Special pulley, minimum diameter and maximum face | Shaft Dimensions, Inches: Standard shaft extension | Shaft Dimensions, Inches: Standard keyway dimensions, L. W. D. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 40 | 220 | 2 | 3450 | 583.5C | ..... | ..... | 455 | 515 | 330 | 12 | ..... | 110 | 1984*c* | 1159*c* | 804 | 80 | ..... | ..... | $2\frac{3}{8}$ | ..... | 5 | $6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$ |
| | | 4 | 1750 | 583C | 440 | 500 | 423 | 483 | 298 | 12 | 5 | 110 | 1124 | 1199 | 804 | 80 | 40 | 10×9 | $2\frac{3}{8}$ | $7\frac{1}{2}\times12$ | 8 | $6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$ |
| | | 6 | 1160 | 587C | 498 | 558 | 479 | 539 | 354 | 12 | 7 | 110 | 1237 | 1312 | 907 | 80 | 50 | 11×10 | $2\frac{3}{8}$ | 10×12 | 8 | $6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$ |
| | | 8 | 870 | 644C | 568 | 628 | 541 | 601 | 416 | 16 | 11 | 115 | 1809 | 1864 | 1369 | 165 | 75 | 12×12 | $2\frac{7}{8}$ | 10×14 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 10 | 690 | 750C | 680 | 740 | 645 | 705 | 520 | 21 | 14 | 120 | 2550 | 2625 | 2090 | 165 | 95 | 14×12 | $3\frac{3}{8}$ | 11×14 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 12 | 575 | 752C | 734 | 794 | 699 | 759 | 574 | 21 | 14 | 120 | 2577 | 2652 | 2117 | 165 | 95 | 14×12 | $3\frac{3}{8}$ | 12×14 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 14 | 490 | 754C | 820 | 880 | 780 | 840 | 655 | 21 | 19 | 120 | 2748 | 2823 | 2158 | 165 | 125 | 16×13 | $3\frac{3}{8}$ | 15×18 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 16 | 435 | 774C | 891 | 951 | 845 | 905 | 720 | 21 | 25 | 120 | 3131 | 3206 | 2596 | 165 | 170 | 18×15 | $3\frac{3}{8}$ | 15×18 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | 440<br>550 | 2 | 3450 | 583.5C | ..... | ..... | 420 | 492 | 330 | 12 | ..... | 110 | 1084*c* | 1159*c* | 804 | 80 | ..... | ..... | $2\frac{3}{8}$ | ..... | 5 | $6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$ |
| | | 4 | 1750 | 583C | 405 | 477 | 388 | 460 | 298 | 12 | 5 | 110 | 1124 | 1199 | 804 | 80 | 40 | 10×9 | $2\frac{3}{8}$ | $7\frac{1}{2}\times12$ | 8 | $6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$ |
| | | 6 | 1160 | 587C | 463 | 535 | 444 | 516 | 354 | 12 | 7 | 110 | 1237 | 1312 | 907 | 80 | 50 | 11×10 | $2\frac{3}{8}$ | 10×12 | 8 | $6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$ |
| | | 8 | 870 | 644C | 533 | 605 | 506 | 578 | 416 | 16 | 11 | 115 | 1809 | 1864 | 1369 | 165 | 75 | 12×12 | $2\frac{7}{8}$ | 10×14 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 10 | 690 | 750C | 645 | 717 | 610 | 682 | 520 | 21 | 14 | 120 | 2559 | 2625 | 2090 | 165 | 95 | 14×12 | $3\frac{3}{8}$ | 11×14 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 12 | 575 | 752C | 699 | 771 | 664 | 736 | 574 | 21 | 14 | 120 | 2570 | 2652 | 2117 | 165 | 95 | 14×12 | $3\frac{3}{8}$ | 12×14 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 14 | 490 | 754C | 785 | 857 | 745 | 817 | 655 | 21 | 19 | 120 | 2748 | 2823 | 2158 | 165 | 125 | 14×12 | $3\frac{3}{8}$ | 15×18 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 16 | 435 | 774C | 856 | 928 | 810 | 882 | 720 | 21 | 25 | 120 | 3131 | 3206 | 2596 | 165 | 170 | 18×15 | $3\frac{3}{8}$ | 15×18 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | 2200 | 4 | 1750 | 583C | 571 | 981 | 554 | 934 | 389 | 12 | 5 | 110 | 1224 | 2424 | 804 | 80 | 40 | 10×9 | $2\frac{3}{8}$ | $7\frac{1}{2}\times12$ | 8 | $6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$ |
| | | 6 | 1160 | 642C | 662 | 1072 | 639 | 1049 | 474 | 16 | 7 | 115 | 1817 | 3017 | 1302 | 165 | 50 | 11×10 | $2\frac{7}{8}$ | 10×12 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 8 | 870 | 652C | 724 | 1134 | 697 | 1107 | 532 | 16 | 11 | 115 | 2089 | 3280 | 1540 | 165 | 75 | 12×12 | $2\frac{7}{8}$ | 10×12 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 10 | 690 | 750C | 845 | 1255 | 810 | 1220 | 645 | 21 | 14 | 120 | 2650 | 3850 | 2090 | 165 | 95 | 14×12 | $3\frac{3}{8}$ | 11×12 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | 220 | 2 | 3450 | 587.5C | ..... | ..... | 520 | 590 | 390 | 12 | ..... | 110 | 1187*c* | 1262*c* | 907 | 80 | ..... | ..... | $2\frac{3}{8}$ | ..... | 5 | $6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$ |
| | | 4 | 1750 | 587C | 496 | 566 | 477 | 547 | 347 | 12 | 7 | 110 | 1237 | 1312 | 907 | 80 | 50 | 11×10 | $2\frac{3}{8}$ | ..... | 8 | $6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$ |
| | | 6 | 1160 | 642C | 567 | 637 | 540 | 610 | 410 | 16 | 11 | 115 | 1742 | 1817 | 1302 | 165 | 75 | 12×12 | $2\frac{7}{8}$ | 19×14 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 8 | 870 | 644C | 639 | 709 | 609 | 679 | 479 | 16 | 14 | 115 | 1829 | 1904 | 1369 | 165 | 95 | 14×12 | $2\frac{7}{8}$ | 11×14 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 10 | 690 | 752C | 757 | 827 | 717 | 787 | 587 | 21 | 19 | 120 | 2607 | 2682 | 2117 | 165 | 125 | 16×13 | $3\frac{3}{8}$ | 12×18 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 12 | 575 | 752C | 820 | 890 | 780 | 850 | 650 | 21 | 19 | 120 | 2607 | 2682 | 2117 | 165 | 125 | 16×13 | $3\frac{3}{8}$ | 12×18 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 14 | 490 | 762C | 918 | 988 | 872 | 942 | 742 | 21 | 25 | 120 | 2849 | 2924 | 2314 | 165 | 170 | 18×15 | $3\frac{3}{8}$ | 15×14 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 16 | 435 | 774C | 991 | 1061 | 945 | 1015 | 815 | 21 | 25 | 120 | 3031 | 3406 | 2596 | 165 | 170 | 18×15 | $3\frac{3}{8}$ | 15×18 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |

| | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 50 | 440 550 | 2 | 3450 | 587.5C | ..... | ..... | 484 | 555 | 390 | 12 | ..... | 110 | 1187*c* | 1262*c* | 907 | 80 | ..... | ..... | $2\frac{3}{8}$ | ..... | 5 | $6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$ |
| | | 4 | 1750 | 587C | 460 | 531 | 441 | 512 | 347 | 12 | 7 | 110 | 1237 | 1312 | 907 | 80 | 50 | 11×10 | $2\frac{3}{8}$ | ..... | 8 | $6\frac{7}{8}\times\frac{9}{16}\times\frac{3}{16}$ |
| | | 6 | 1160 | 642C | 531 | 602 | 504 | 575 | 410 | 16 | 11 | 115 | 1742 | 1817 | 1302 | 165 | 75 | 12×12 | $2\frac{7}{8}$ | 10×14 | $9\frac{1}{2}$ | $8\frac{1}{8}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 8 | 870 | 644C | 603 | 674 | 573 | 644 | 479 | 16 | 14 | 115 | 1829 | 1904 | 1369 | 165 | 95 | 14×12 | $2\frac{7}{8}$ | 11×14 | $9\frac{1}{2}$ | $8\frac{1}{8}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 10 | 690 | 752C | 721 | 792 | 681 | 752 | 587 | 21 | 19 | 120 | 2607 | 2682 | 2117 | 165 | 125 | 16×13 | $3\frac{3}{8}$ | 12×18 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 12 | 575 | 752C | 784 | 855 | 744 | 815 | 650 | 21 | 19 | 120 | 2607 | 2682 | 2117 | 165 | 125 | 16×13 | $3\frac{3}{8}$ | 12×18 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 14 | 490 | 762C | 882 | 953 | 836 | 907 | 742 | 21 | 25 | 120 | 2849 | 2924 | 2314 | 165 | 170 | 18×15 | $3\frac{3}{8}$ | 15×14 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 16 | 435 | 774C | 955 | 1026 | 909 | 980 | 815 | 21 | 25 | 120 | 3131 | 3406 | 2596 | 165 | 170 | 18×15 | $3\frac{3}{8}$ | 15×18 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | 2200 | 4 | 1750 | 643C | 629 | 1034 | 606 | 1011 | 436 | 16 | 7 | 115 | 1929 | 3129 | 1369 | 165 | 50 | 11×10 | $2\frac{7}{8}$ | ..... | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 6 | 1160 | 646C | 729 | 1134 | 702 | 1107 | 532 | 16 | 11 | 125 | 2058 | 3258 | 1518 | 165 | 75 | 12×12 | $2\frac{7}{8}$ | 11×14 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 8 | 870 | 752C | 793 | 1198 | 758 | 1163 | 588 | 21 | 14 | 120 | 2577 | 3877 | 2117 | 165 | 95 | 14×12 | $3\frac{3}{8}$ | 12×14 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 10 | 690 | 752C | 912 | 1317 | 872 | 1277 | 702 | 21 | 19 | 120 | 2807 | 4007 | 2117 | 165 | 125 | 16×13 | $3\frac{3}{8}$ | 14×15 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 12 | 575 | 774C | 964 | 1369 | 924 | 1329 | 754 | 21 | 19 | 110 | 3086 | 4286 | 2596 | 165 | 125 | 16×13 | $3\frac{3}{8}$ | 15×18 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | 220 | 4 | 1750 | 645C | ..... | ..... | 521 | 843 | 393 | 16 | ..... | 115 | 2155*c* | 2455*c* | 1590 | 165 | ..... | ..... | $2\frac{7}{8}$ | ..... | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 6 | 1160 | 644C | 616 | 938 | 589 | 911 | 461 | 16 | 11 | 115 | 2009 | 2309 | 1369 | 165 | 75 | 12×12 | $2\frac{7}{8}$ | 11×14 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 8 | 870 | 662C | 697 | 1019 | 667 | 989 | 539 | 16 | 14 | 115 | 2377 | 2677 | 1717 | 165 | 95 | 14×12 | $2\frac{7}{8}$ | 12×14 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 10 | 690 | 752C | 818 | 1140 | 778 | 1100 | 650 | 21 | 19 | 120 | 2817 | 3117 | 2127 | 165 | 125 | 16×13 | $3\frac{3}{8}$ | 14×15 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 12 | 575 | 774C | 895 | 1217 | 849 | 1171 | 721 | 21 | 25 | 120 | 3331 | 3631 | 2596 | 165 | 170 | 18×15 | $3\frac{3}{8}$ | 15×18 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 14 | 490 | 856C | 1022 | 1344 | 953 | 1275 | 825 | 44 | 25 | 230 | 4244 | 3544 | 3341 | 330 | 170 | 18×15 | $3\frac{7}{8}$ | 16×20 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 16 | 435 | 856C | 1109 | 1431 | 1038 | 1360 | 910 | 44 | 27 | 230 | 4244 | 3544 | 3344 | 330 | 170 | 20×15 | $3\frac{7}{8}$ | 16×20 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| 60 | 440 550 | 4 | 1750 | 645C | ..... | ..... | 521 | 649 | 393 | 16 | ..... | 115 | 2105*c* | 2155*c* | 1590 | 165 | ..... | ..... | $2\frac{7}{8}$ | ..... | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 6 | 1160 | 644C | 616 | 744 | 589 | 717 | 461 | 16 | 11 | 115 | 1959 | 2009 | 1369 | 165 | 75 | 12×12 | $2\frac{7}{8}$ | 11×14 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 8 | 870 | 662C | 697 | 825 | 667 | 795 | 539 | 16 | 14 | 115 | 2327 | 2377 | 1717 | 165 | 95 | 14×12 | $2\frac{7}{8}$ | 12×14 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 10 | 690 | 752C | 818 | 946 | 778 | 906 | 650 | 21 | 19 | 120 | 2767 | 2817 | 2127 | 165 | 125 | 16×13 | $3\frac{3}{8}$ | 14×15 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 12 | 575 | 774C | 895 | 1023 | 849 | 977 | 721 | 21 | 25 | 120 | 3281 | 3331 | 2596 | 165 | 170 | 18×15 | $3\frac{3}{8}$ | 15×18 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 14 | 490 | 856C | 1022 | 1150 | 953 | 1081 | 825 | 44 | 25 | 230 | 4194 | 4244 | 3341 | 330 | 170 | 18×15 | $3\frac{7}{8}$ | 16×20 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 16 | 435 | 856C | 1109 | 1237 | 1038 | 1166 | 910 | 44 | 27 | 230 | 4194 | 4244 | 3344 | 330 | 170 | 20×15 | $3\frac{7}{8}$ | 16×20 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | 2200 | 4 | 1750 | 653C | ..... | ..... | 648 | 1029 | 482 | 16 | ..... | 115 | 2152*c* | 3277*c* | 1562 | 165 | ..... | ..... | $2\frac{7}{8}$ | ..... | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 6 | 1160 | 664C | 777 | 1158 | 750 | 1131 | 584 | 16 | 11 | 115 | 2425 | 3558 | 1768 | 165 | 75 | 12×12 | $2\frac{7}{8}$ | 11×14 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 8 | 870 | 762C | 846 | 1227 | 811 | 1192 | 645 | 21 | 14 | 120 | 2999 | 4124 | 2314 | 165 | 95 | 14×12 | $3\frac{3}{8}$ | 12×14 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 10 | 690 | 762C | 960 | 1341 | 920 | 1301 | 754 | 21 | 19 | 120 | 3029 | 4154 | 2314 | 165 | 125 | 16×13 | $3\frac{3}{8}$ | 12×15 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 12 | 575 | 854C | 1046 | 1427 | 977 | 1358 | 811 | 44 | 25 | 120 | 4203 | 5328 | 3278 | 330 | 170 | 18×15 | $3\frac{3}{8}$ | 15×18 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | 220 | 4 | 1750 | 645C | ..... | ..... | 594 | 936 | 461 | 16 | .... | 115 | 2055*c* | 2355*c* | 1490 | 165 | ..... | ..... | $2\frac{7}{8}$ | ..... | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 6 | 1160 | 662C | 700 | 1042 | 670 | 1012 | 537 | 16 | 14 | 115 | 2177 | 2677 | 1717 | 165 | 95 | 14×12 | $2\frac{7}{8}$ | 13×14 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 8 | 870 | 752C | 799 | 1141 | 759 | 1101 | 626 | 21 | 19 | 120 | 2807 | 3107 | 2117 | 165 | 125 | 16×13 | $3\frac{3}{8}$ | 14×16 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 10 | 690 | 754C | 913 | 1255 | 867 | 1209 | 734 | 21 | 25 | 120 | 2893 | 3193 | 2158 | 165 | 170 | 18×15 | $3\frac{3}{8}$ | 15×18 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 12 | 575 | 774C | 995 | 1337 | 949 | 1291 | 816 | 21 | 25 | 120 | 3331 | 3631 | 2596 | 165 | 170 | 18×15 | $3\frac{3}{8}$ | 16×20 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 14 | 490 | 854C | 1140 | 1482 | 1069 | 1411 | 936 | 44 | 27 | 230 | 4244 | 4544 | 3344 | 330 | 170 | 20×15 | $3\frac{7}{8}$ | 16×20 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 16 | 435 | 874C | 1241 | 1583 | 1168 | 1510 | 1035 | 44 | 29 | 230 | 4546 | 4846 | 3586 | 330 | 230 | 21×17 | $3\frac{7}{8}$ | 17×20 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| 75 | 440 550 | 4 | 1750 | 645C | ..... | ..... | 594 | 717 | 461 | 16 | ..... | 115 | 2005*c* | 2055*c* | 1490 | 165 | ..... | ..... | $2\frac{7}{8}$ | ..... | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 6 | 1160 | 662C | 700 | 823 | 670 | 793 | 537 | 16 | 14 | 115 | 2327 | 2377 | 1717 | 165 | 95 | 14×12 | $2\frac{7}{8}$ | 13×14 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 8 | 870 | 752C | 799 | 922 | 759 | 882 | 626 | 21 | 19 | 120 | 2757 | 2807 | 2117 | 165 | 125 | 16×13 | $3\frac{3}{8}$ | 14×16 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 10 | 690 | 754C | 913 | 1036 | 867 | 990 | 734 | 21 | 25 | 120 | 2843 | 2893 | 2158 | 165 | 170 | 18×15 | $3\frac{3}{8}$ | 15×18 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 12 | 575 | 774C | 995 | 1118 | 949 | 1072 | 816 | 21 | 25 | 120 | 3281 | 3331 | 2596 | 165 | 170 | 18×15 | $3\frac{3}{8}$ | 16×20 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 14 | 490 | 854C | 1136 | 1259 | 1065 | 1188 | 936 | 44 | 27 | 230 | 4194 | 4244 | 3344 | 330 | 170 | 20×15 | $3\frac{7}{8}$ | 16×20 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 16 | 435 | 874C | 1231 | 1354 | 1158 | 1281 | 1035 | 44 | 29 | 230 | 4496 | 4546 | 3586 | 330 | 230 | 21×17 | $3\frac{7}{8}$ | 17×20 | $13\frac{1}{2}$ | $12\times\frac{13}{16}\times\frac{5}{16}$ |

For notes and explanation of reference letters, see last page of this table.

TABLE 152—*Continued*

| Horse-power | Volts | Number of Poles | Full Load, R.p.m. | Frame | Price, Dollars: Complete motor with: Type "A" manual starter Class 10,700 | Price, Dollars: Complete motor with: Magnetic auto-transformer type starter | Price, Dollars: Bare motor with: Type "A" manual starter Class 10,700 | Price, Dollars: Bare motor with: Magnetic auto-transformer type starter | Price, Dollars: Bare motor | Price, Dollars: Omission or addition for: Rails | Price, Dollars: Omission or addition for: Pulley | Price, Dollars: Addition for vertical parts | Approximate Shipping Weight, Pounds: Complete motor with: Type A starter | Approximate Shipping Weight, Pounds: Complete motor with: Magnetic starter | Approximate Shipping Weight, Pounds: Bare motor | Approximate Shipping Weight, Pounds: Omission or addition for: Rails | Approximate Shipping Weight, Pounds: Omission or addition for: Pulley | Pulley Dimensions, Inches: Standard diameter and face | Pulley Dimensions, Inches: Bore | Pulley Dimensions, Inches: Special pulley, minimum diameter and maximum face | Shaft Dimensions, Inches: Standard shaft extension | Shaft Dimensions, Inches: Standard keyway dimensions, L. W. D. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 75 | 2200 | 4 | 1750 | 653C | ..... | ..... | 718 | 1098 | 547 | 16 | ..... | 115 | 2252*c* | 3377*c* | 1562 | 165 | ..... | ..... | $2\frac{7}{8}$ | ..... | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 6 | 1160 | 664C | 860 | 1240 | 830 | 1210 | 659 | 16 | 14 | 115 | 2453 | 3578 | 1768 | 165 | 95 | 14×12 | $2\frac{7}{8}$ | 13×14 | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 8 | 870 | 762C | 934 | 1314 | 894 | 1274 | 723 | 21 | 19 | 120 | 3029 | 4154 | 2314 | 165 | 125 | 16×13 | $3\frac{3}{8}$ | 15×18 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 10 | 690 | 774C | 1043 | 1423 | 997 | 1377 | 826 | 21 | 25 | 120 | 3356 | 4481 | 2596 | 165 | 170 | 18×15 | $3\frac{3}{8}$ | 16×20 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 12 | 575 | 854C | 1135 | 1515 | 1066 | 1446 | 895 | 44 | 25 | 230 | 4203 | 4328 | 3278 | 330 | 170 | 18×15 | $3\frac{7}{8}$ | 16×20 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | 220 | 4 | 1750 | 663C | ..... | ..... | 764 | 1054 | 574 | 16 | ..... | 115 | 2334*c* | 2634*c* | 1769 | 165 | ..... | ..... | $2\frac{7}{8}$ | ..... | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 6 | 1160 | 752C | 902 | 1192 | 862 | 1152 | 672 | 21 | 19 | 120 | 2807 | 3107 | 2117 | 165 | 125 | 16×13 | $3\frac{3}{8}$ | 13×14 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 8 | 870 | 754C | 997 | 1287 | 951 | 1241 | 761 | 21 | 25 | 120 | 2893 | 3193 | 2158 | 165 | 170 | 18×15 | $3\frac{3}{8}$ | 14×16 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 10 | 590 | 854C | 1129 | 1419 | 1058 | 1348 | 868 | 44 | 27 | 230 | 4178 | 4478 | 3278 | 330 | 170 | 20×15 | $3\frac{7}{8}$ | 15×18 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 12 | 575 | 854C | 1226 | 1516 | 1153 | 1443 | 963 | 44 | 29 | 230 | 4238 | 4538 | 3278 | 330 | 230 | 21×17 | $3\frac{7}{8}$ | 16×20 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 14 | 490 | 876C | 1373 | 1663 | 1290 | 1580 | 1100 | 44 | 39 | 230 | 4910 | 5210 | 3740 | 330 | 440 | 25×17 | $3\frac{7}{8}$ | 22×22 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 16 | 435 | 936 | 1492 | 1782 | 1410 | 1700 | 1220 | 53 | 29 | ..... | 5880 | 5580 | 4070 | 880 | 230 | 21×17 | $3\frac{7}{8}$ | 20×20 | $13\frac{1}{2}$ | $11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$ |
| 100 | 440 550 | 4 | 1750 | 663C | ..... | ..... | 712 | 830 | 574 | 16 | ..... | 115 | 2284*c* | 2334*c* | 1769 | 165 | ..... | ..... | $2\frac{7}{8}$ | ..... | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 6 | 1160 | 752C | 850 | 968 | 810 | 928 | 672 | 21 | 19 | 120 | 2757 | 2807 | 2117 | 165 | 125 | 16×13 | $3\frac{3}{8}$ | 13×14 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 8 | 870 | 754C | 945 | 1063 | 899 | 1017 | 761 | 21 | 25 | 120 | 2843 | 2893 | 2158 | 165 | 170 | 18×15 | $3\frac{3}{8}$ | 14×16 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 10 | 690 | 854C | 1077 | 1195 | 1006 | 1124 | 868 | 44 | 27 | 230 | 4128 | 4178 | 3278 | 330 | 170 | 20×15 | $3\frac{7}{8}$ | 15×18 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 12 | 575 | 854C | 1174 | 1292 | 1101 | 1219 | 963 | 44 | 29 | 230 | 4188 | 4238 | 3278 | 330 | 230 | 21×17 | $3\frac{7}{8}$ | 16×20 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 14 | 490 | 876C | 1321 | 1439 | 1238 | 1356 | 1100 | 44 | 39 | 230 | 4860 | 4910 | 3740 | 330 | 440 | 25×17 | $3\frac{7}{8}$ | 22×22 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 16 | 435 | 936 | 1430 | 1558 | 1358 | 1476 | 1220 | 53 | 29 | ..... | 5880 | 5580 | 4070 | 880 | 230 | 21×17 | $3\frac{7}{8}$ | 20×20 | $13\frac{1}{2}$ | $11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$ |
| | 2200 | 4 | 1750 | 663C | ..... | ..... | 829 | 1213 | 653 | 16 | ..... | 115 | 1959*c* | 3484*c* | 1769 | 165 | ..... | ..... | $2\frac{7}{8}$ | ..... | $9\frac{1}{2}$ | $8\frac{1}{4}\times\frac{11}{16}\times\frac{1}{4}$ |
| | | 6 | 1160 | 760C | 993 | 1377 | 953 | 1337 | 777 | 21 | 19 | 120 | 2915 | 4040 | 2200 | 165 | 125 | 16×13 | $3\frac{3}{8}$ | 13×16 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 8 | 870 | 774C | 1069 | 1453 | 1023 | 1407 | 847 | 21 | 25 | 120 | 3356 | 4481 | 2596 | 165 | 170 | 18×15 | $3\frac{3}{8}$ | 16×18 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 10 | 690 | 856C | 1183 | 1567 | 1112 | 1496 | 936 | 44 | 27 | 230 | 4269 | 5394 | 3344 | 330 | 170 | 20×15 | $3\frac{7}{8}$ | 17×20 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 12 | 575 | 874C | 1273 | 1657 | 1200 | 1584 | 1024 | 44 | 29 | 230 | 4571 | 5696 | 3586 | 330 | 230 | 21×17 | $3\frac{7}{8}$ | 19×20 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 14 | 490 | 938 | 1398 | 1782 | 1316 | 1700 | 1140 | 53 | 29 | ..... | 6928 | 5803 | 4268 | 880 | 230 | 21×17 | $3\frac{7}{8}$ | 20×20 | $13\frac{1}{2}$ | $11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 16 | 435 | 938 | 1518 | 1902 | 1436 | 1820 | 1260 | 53 | 29 | ..... | 6928 | 5803 | 4268 | 880 | 230 | 21×17 | $3\frac{7}{8}$ | 20×20 | $13\frac{1}{2}$ | $11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$ |

| | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 4 | 1750 | 761C | ..... | ..... | 880 | 1388 | 680 | 21 | ..... | 120 | 3883*c* | 3983*c* | 2468 | 165 | ..... | ..... | $3\frac{3}{8}$ | ..... | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 6 | 1760 | 764C | ..... | ..... | 990 | 1498 | 790 | 21 | ..... | 120 | 3915*c* | 4015*c* | 2500 | 165 | ..... | ..... | $3\frac{3}{8}$ | ..... | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 8 | 870 | 774C | 1141 | 1649 | 1093 | 1601 | 893 | 21 | 27 | 120 | 4181 | 4281 | 2596 | 165 | 170 | 20×15 | $3\frac{3}{8}$ | 18×18 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | 220 | 10 | 690 | 856C | 1269 | 1777 | 1186 | 1694 | 986 | 44 | 39 | 230 | 5364 | 5464 | 3344 | 330 | 440 | 25×17 | $3\frac{7}{8}$ | 20×20 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 12 | 575 | 874C | 1381 | 1889 | 1298 | 1806 | 1098 | 44 | 39 | 230 | 5696 | 5706 | 3586 | 330 | 440 | 25×17 | $3\frac{7}{8}$ | 20×20 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 14 | 490 | 936 | 1510 | 2018 | 1420 | 1928 | 1220 | 53 | 37 | ..... | 6430 | 6530 | 4970 | 880 | 230 | 23×19 | $3\frac{7}{8}$ | 22×20 | $13\frac{1}{2}$ | $11\frac{3}{4}\times\frac{13}{16}\times\frac{5}{16}$ |
| | | 16 | 435 | 938 | 1634 | 2142 | 1540 | 2048 | 1340 | 53 | 41 | ..... | 6838 | 6938 | 4268 | 880 | 440 | 27×17 | $3\frac{7}{8}$ | 24×20 | $13\frac{1}{2}$ | $11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 4 | 1750 | 761C | ..... | ..... | 823 | 1170 | 680 | 21 | .... | 120 | 3033*c* | 3333*c* | 2468 | 165 | ..... | ..... | $3\frac{3}{8}$ | ..... | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 6 | 1160 | 764C | ..... | ..... | 933 | 1280 | 790 | 21 | ..... | 120 | 3065*c* | 3365*c* | 2500 | 165 | ..... | ..... | $3\frac{3}{8}$ | ..... | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 8 | 870 | 774C | 1084 | 1431 | 1036 | 1383 | 893 | 21 | 27 | 120 | 3331 | 3631 | 2596 | 165 | 170 | 20×15 | $3\frac{3}{8}$ | 18×18 | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| 125 | 440<br>550 | 10 | 690 | 856C | 1212 | 1559 | 1129 | 1476 | 986 | 44 | 39 | 230 | 4514 | 4814 | 3341 | 330 | 440 | 25×17 | $3\frac{7}{8}$ | 20×20 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 12 | 575 | 874C | 1324 | 1617 | 1241 | 1588 | 1098 | 44 | 39 | 230 | 4756 | 5056 | 3586 | 330 | 440 | 25×17 | $3\frac{7}{8}$ | 20×20 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 14 | 490 | 936 | 1453 | 1800 | 1363 | 1710 | 1220 | 53 | 37 | ..... | 5580 | 5880 | 4070 | 880 | 230 | 23×19 | $3\frac{7}{8}$ | 22×20 | $13\frac{1}{2}$ | $11\frac{3}{4}\times\frac{13}{16}\times\frac{5}{16}$ |
| | | 16 | 435 | 938 | 1577 | 1924 | 1483 | 1830 | 1340 | 53 | 41 | ..... | 5988 | 6288 | 4268 | 880 | 440 | 27×17 | $3\frac{7}{8}$ | 24×20 | $13\frac{1}{2}$ | $11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 4 | 1750 | 761C | ..... | ..... | 942 | 1327 | 752 | 21 | ..... | 120 | 3108*c* | 4233*c* | 2468 | 165 | ..... | ..... | $3\frac{3}{8}$ | ..... | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 6 | 1160 | 772C | ..... | ..... | 1080 | 1465 | 890 | 21 | ..... | 120 | 3072*c* | 4197*c* | 2452 | 165 | ..... | ..... | $3\frac{3}{8}$ | ..... | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 8 | 870 | 856C | 1228 | 1613 | 1157 | 1542 | 967 | 44 | 27 | 230 | 3319 | 4444 | 3344 | 330 | 170 | 20×15 | $3\frac{7}{8}$ | 16×20 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | 2200 | 10 | 690 | 876C | 1306 | 1691 | 1233 | 1618 | 1043 | 44 | 29 | 230 | 4775 | 5990 | 3740 | 330 | 230 | 21×17 | $3\frac{7}{8}$ | 20×20 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 12 | 575 | 938 | 1429 | 1814 | 1337 | 1722 | 1147 | 53 | 39 | 275 | 5063 | 7188 | 4268 | 880 | 440 | 25×17 | $3\frac{7}{8}$ | 20×20 | $13\frac{1}{2}$ | $11\frac{3}{4}\times\frac{13}{16}\times\frac{5}{16}$ |
| | | 14 | 490 | 938 | 1550 | 1935 | 1460 | 1845 | 1270 | 53 | 37 | ..... | 5853 | 6978 | 4268 | 880 | 230 | 23×19 | $3\frac{7}{8}$ | 22×20 | $13\frac{1}{2}$ | $11\frac{3}{4}\times\frac{13}{16}\times\frac{5}{16}$ |
| | | 16 | 435 | 1000 | 1684 | 2069 | 1580 | 1965 | 1390 | 63 | 41 | ..... | 6795 | 7920 | 5830 | 1050 | 440 | 27×17 | $3\frac{7}{8}$ | 24×20 | $13\frac{1}{2}$ | $11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 4 | 1750 | 771.1C | ..... | ..... | 1163 | 1586 | 783 | 21 | ..... | 120 | 3850*c* | 4000*c* | 2435 | 165 | ..... | ..... | $3\frac{3}{8}$ | ..... | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 6 | 1160 | 772C | ..... | ..... | 1298 | 1721 | 918 | 21 | ..... | 120 | 3847*c* | 3997*c* | 2432 | 165 | ..... | ..... | $3\frac{3}{8}$ | ..... | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 8 | 870 | 854C | 1478 | 1901 | 1405 | 1828 | 1025 | 44 | 29 | 230 | 5088 | 5238 | 3278 | 330 | 230 | 21×17 | $3\frac{7}{8}$ | 19×20 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | 220 | 10 | 690 | 874C | 1561 | 1984 | 1478 | 1901 | 1098 | 44 | 39 | 230 | 5696 | 5756 | 3586 | 330 | 440 | 25×17 | $3\frac{7}{8}$ | 22×20 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 12 | 575 | 876C | 1685 | 2108 | 1602 | 2025 | 1222 | 44 | 39 | 230 | 5760 | 5910 | 3740 | 330 | 440 | 25×17 | $3\frac{7}{8}$ | 22×22 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 14 | 490 | 938 | 1813 | 2236 | 1720 | 2143 | 1340 | 53 | 40 | ..... | 6838 | 6988 | 4268 | 880 | 440 | 28×17 | $3\frac{7}{8}$ | 25×20 | $13\frac{1}{2}$ | $11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 16 | 435 | 1000 | 1943 | 2366 | 1840 | 2263 | 1460 | 63 | 40 | ..... | 8570 | 8720 | 5830 | 1050 | 440 | 28×20 | $4\frac{1}{8}$ | 25×24 | 16 | $15\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$ |
| | | 4 | 1750 | 771.1C | ..... | ..... | 930 | 1283 | 783 | 21 | ..... | 120 | 3300*c* | 3300*c* | 2435 | 165 | ..... | ..... | $3\frac{3}{8}$ | ..... | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 6 | 1160 | 772C | ..... | ..... | 1065 | 1418 | 918 | 21 | ..... | 120 | 2997*c* | 3297*c* | 2432 | 165 | ..... | ..... | $3\frac{3}{8}$ | ..... | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 8 | 870 | 854C | 1245 | 1598 | 1172 | 1525 | 1025 | 44 | 29 | 230 | 4238 | 4538 | 3278 | 330 | 230 | 21×17 | $3\frac{7}{8}$ | 19×20 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| 150 | 440<br>550 | 10 | 690 | 874C | 1328 | 1681 | 1245 | 1598 | 1098 | 44 | 39 | 230 | 4756 | 5056 | 3586 | 330 | 440 | 25×17 | $3\frac{7}{8}$ | 22×20 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 12 | 575 | 876C | 1452 | 1805 | 1369 | 1722 | 1222 | 44 | 39 | 230 | 4910 | 5210 | 3740 | 330 | 440 | 25×17 | $3\frac{7}{8}$ | 22×22 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 14 | 490 | 938 | 1580 | 1933 | 1487 | 1840 | 1340 | 53 | 40 | ..... | 4988 | 6288 | 4268 | 880 | 449 | 28×17 | $3\frac{7}{8}$ | 25×20 | $13\frac{1}{2}$ | $11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 16 | 435 | 1000 | 1710 | 2063 | 1607 | 1960 | 1460 | 63 | 40 | ..... | 7720 | 8020 | 5830 | 1050 | 440 | 28×20 | $4\frac{1}{8}$ | 25×24 | 16 | $15\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$ |
| | | 4 | 1750 | 771.1C | ..... | ..... | 1052 | 1427 | 852 | 21 | ..... | 120 | 3075*c* | 4300*c* | 2435 | 165 | ..... | ..... | $3\frac{3}{8}$ | ..... | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 6 | 1160 | 772C | ..... | ..... | 1204 | 1579 | 1004 | 21 | ..... | 120 | 3072 | 4297*c* | 2432 | 165 | ..... | ..... | $3\frac{3}{8}$ | ..... | $10\frac{1}{4}$ | $8\frac{7}{8}\times\frac{13}{16}\times\frac{9}{32}$ |
| | | 8 | 870 | 870C | 1358 | 1733 | 1285 | 1660 | 1085 | 44 | 29 | 230 | 4497 | 5722 | 3462 | 330 | 230 | 21×17 | $3\frac{7}{8}$ | 19×20 | $13\frac{1}{2}$ | $12\times\frac{15}{16}\times\frac{5}{16}$ |
| | 2200 | 10 | 690 | 876C | 1430 | 1805 | 1347 | 1722 | 1147 | 44 | 39 | 275 | 4985 | 6210 | 3940 | 330 | 440 | 25×17 | $3\frac{7}{8}$ | 22×20 | $13\frac{1}{2}$ | $11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 12 | 575 | 954 | 1554 | 1929 | 1467 | 1842 | 1267 | 53 | 34 | 275 | 6388 | 7613 | 4593 | 880 | 440 | 25×17 | $3\frac{7}{8}$ | 22×22 | $13\frac{1}{2}$ | $11\frac{3}{4}\times\frac{15}{16}\times\frac{5}{16}$ |
| | | 14 | 490 | 954A | 1678 | 2053 | 1590 | 1965 | 1390 | 53 | 35 | ..... | 6388 | 7613 | 4593 | 880 | 440 | 27×19 | $4\frac{3}{8}$ | 22×23 | $15\frac{1}{2}$ | $13\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$ |
| | | 16 | 435 | 1000 | 1813 | 2188 | 1710 | 2085 | 1510 | 63 | 40 | ..... | 7795 | 9020 | 5830 | 1050 | 440 | 28×20 | $4\frac{7}{8}$ | 25×24 | 16 | $15\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$ |

For notes and explanation of reference letters, see last page of this table.

TABLE 152—*Concluded*

| Horse-power | Volts | Number of Poles | Full Load, R.p.m. | Frame | Price, Dollars | | | | | | | Approximate Shipping Weight, Pounds | | | | | Pulley Dimensions, Inches | | | Shaft Dimensions, Inches | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | Complete motor with | | Bare motor with | | | Omission or addition for | | | Complete motor with | | | Omission or addition for | | | | | | |
| | | | | | Type "A" manual starter Class 10,700 | Magnetic auto-transformer type starter | Type "A" manual starter Class 10,700 | Magnetic auto-transformer type starter | Bare motor | Rails | Pulley | Addition for vertical parts | Type A starter | Magnetic starter | Bare motor | Rails | Pulley | Standard diameter and face | Bore | Special pulley, minimum diameter and maximum face | Standard shaft extension | Standard keyway dimensions, L. W. D. |
| | | 4 | 1750 | 855C | ..... | ..... | 1755 | 2000 | 985 | 44 | ..... | 230 | 5005*c* | 5005*c* | 3225 | 330 | ..... | ..... | $3\frac{7}{8}$ | ..... | $13\frac{1}{2}$ | $12\times\frac{13}{16}\times\frac{5}{16}$ |
| | | 6 | 1160 | 871C | ..... | ..... | 1952 | 2197 | 1182 | 44 | ..... | 230 | 5300*c* | 5300*c* | 3520 | 330 | ..... | ..... | $3\frac{7}{8}$ | ..... | $13\frac{1}{2}$ | $12\times\frac{13}{16}\times\frac{5}{16}$ |
| | | 8 | 870 | 874C | ..... | ..... | 2044 | 2289 | 1274 | 44 | ..... | 230 | 5366*c* | 5366*c* | 3586 | 330 | ..... | ..... | $3\frac{7}{8}$ | ..... | $13\frac{1}{2}$ | $12\times\frac{13}{16}\times\frac{5}{16}$ |
| | 220 | 10 | 690 | 954A | 2161 | 2406 | 2070 | 2315 | 1300 | 53 | 38 | 275 | 7363 | 7363 | 4593 | 880 | 440 | $28\times18$ | $4\frac{3}{8}$ | $23\times22$ | $15\frac{1}{4}$ | $13\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$ |
| | | 12 | 575 | 954A | 2312 | 2557 | 2219 | 2464 | 1449 | 53 | 40 | 275 | 7363 | 7363 | 4593 | 880 | 440 | $32\times20$ | $4\frac{3}{8}$ | $25\times24$ | $15\frac{1}{4}$ | $13\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$ |
| | | 14 | 490 | 1000 | 2458 | 2703 | 2345 | 2590 | 1575 | 63 | 50 | ..... | 8770 | 8770 | 5830 | 1059 | 440 | $33\times20$ | $4\frac{7}{8}$ | $28\times24$ | 16 | $15\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$ |
| | | 16 | 435 | 1000 | 2593 | 2838 | 2470 | 2715 | 1700 | 63 | 60 | ..... | 8960 | 8960 | 5830 | 1059 | 630 | $38\times20$ | $4\frac{7}{8}$ | $32\times24$ | 16 | $15\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$ |
| | | 4 | 1750 | 871C | ..... | ..... | 1171 | 1494 | 985 | 44 | ..... | 230 | 3955*c* | 4255*c* | 3225 | 330 | ..... | ..... | $3\frac{7}{8}$ | ..... | $13\frac{1}{2}$ | $12\times\frac{13}{16}\times\frac{5}{16}$ |
| | | 6 | 1160 | 873C | ..... | ..... | 1368 | 1691 | 1182 | 44 | ..... | 230 | 4250*c* | 4550*c* | 3520 | 330 | ..... | ..... | $3\frac{7}{8}$ | ..... | $13\frac{1}{2}$ | $12\times\frac{13}{16}\times\frac{5}{16}$ |
| | | 8 | 870 | 874C | ..... | ..... | 1460 | 1783 | 1274 | 44 | ..... | 230 | 4316*c* | 4616*c* | 3586 | 330 | ..... | ..... | $3\frac{7}{8}$ | ..... | $13\frac{1}{2}$ | $12\times\frac{13}{16}\times\frac{5}{16}$ |
| 200 | 440 550 | 10 | 690 | 954A | 1577 | 1900 | 1486 | 1809 | 1300 | 53 | 38 | 275 | 6313 | 6613 | 4593 | 880 | 440 | $28\times18$ | $4\frac{3}{8}$ | $23\times22$ | $15\frac{1}{4}$ | $13\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$ |
| | | 12 | 575 | 954A | 1728 | 2051 | 1635 | 1958 | 1449 | 53 | 40 | 275 | 6313 | 6613 | 4593 | 880 | 440 | $32\times20$ | $4\frac{3}{8}$ | $25\times24$ | $15\frac{1}{4}$ | $13\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$ |
| | | 14 | 490 | 1000 | 1874 | 2197 | 1761 | 2084 | 1575 | 63 | 50 | ..... | 7720 | 8020 | 5830 | 1050 | 440 | $33\times20$ | $4\frac{7}{8}$ | $28\times24$ | 16 | $15\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$ |
| | | 16 | 435 | 1002 | 2009 | 2332 | 1886 | 2209 | 1700 | 63 | 60 | ..... | 7910 | 8210 | 5830 | 1050 | 630 | $38\times20$ | $4\frac{7}{8}$ | $32\times24$ | 16 | $15\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$ |
| | | 4 | 1750 | 871C | ..... | ..... | 1242 | 1627 | 1033 | 44 | ..... | 230 | 4391*c* | 5616*c* | 3586 | 330 | ..... | ..... | $3\frac{7}{8}$ | ..... | $13\frac{1}{2}$ | $12\times\frac{13}{16}\times\frac{5}{16}$ |
| | | 6 | 1160 | 873C | ..... | ..... | 1440 | 1825 | 1231 | 44 | ..... | 230 | 4435*c* | 5660*c* | 3630 | 330 | ..... | ..... | $3\frac{7}{8}$ | ..... | $13\frac{1}{2}$ | $12\times\frac{13}{16}\times\frac{5}{16}$ |
| | | 8 | 870 | 939A | 1608 | 1993 | 1522 | 1907 | 1313 | 53 | 33 | 275 | 6305 | 7530 | 4510 | 880 | 440 | $22\times20$ | $4\frac{3}{8}$ | $20\times22$ | $15\frac{1}{4}$ | $13\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$ |
| | 2200 | 10 | 690 | 956A | 1646 | 2031 | 1548 | 1933 | 1339 | 53 | 45 | 275 | 6470 | 7695 | 4672 | 880 | 440 | $28\times18$ | $4\frac{3}{8}$ | $23\times22$ | $15\frac{1}{4}$ | $13\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$ |
| | | 12 | 575 | 1002 | 1817 | 2202 | 1695 | 2080 | 1486 | 63 | 59 | 460 | 8095 | 9320 | 5940 | 1059 | 630 | $32\times20$ | $4\frac{7}{8}$ | $25\times24$ | 16 | $15\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$ |
| | | 14 | 490 | 1000 | 1947 | 2332 | 1819 | 2204 | 1610 | 63 | 65 | ..... | 7985 | 9210 | 5830 | 1050 | 630 | $33\times20$ | $4\frac{7}{8}$ | $28\times24$ | 16 | $15\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$ |
| | | 16 | 435 | 1002 | 2072 | 2457 | 1939 | 2324 | 1730 | 63 | 70 | ..... | 8165 | 9390 | 5940 | 1050 | 700 | $38\times20$ | $4\frac{7}{8}$ | $32\times24$ | 16 | $15\frac{1}{2}\times1\frac{1}{4}\times\frac{7}{16}$ |

NOTES AND EXPLANATION OF REFERENCE LETTERS

*a* Class 200 and 300 frames are supplied with bedplate instead of rails.

*b* Pulleys require special offset hub in most cases, if wider pulley is used, special shaft extension is required.

*c* Pulleys are not included in weight.

*d* Length of shaft extension, pulley end.

*e* Not recommended for belted service.

*f* Price includes a class 10,700, type A enclosed auto-starter providing overload and low voltage protection by automatically resetting relays.

*g* (220, 440 and 550 volts). Price includes a class 11,500, type AF enclosed magnetic auto-starter, "start" and "stop" push button station and "reset" push button. This starter provides overload protection and low-voltage release or protection. When ordering push button station specify whether low-voltage protection or low-voltage release is desired. The "start" and "stop" push button station may be omitted at $5.00 if another type of master switch is desired.

*h* (2200 volts). Price includes a Class 11,800 magnetic starter and "start" and "stop" push button station. This starter provides overload protection and low-voltage release or protection. When ordering push button station specify whether low-voltage protection or low-voltage release is desired. "Start" and "stop" push button station may be omitted at $2.50 if another type of master switch is desired.

*i* (220 volts). Price includes a Class 11,550 enclosed magnetic starter and "start" and "stop" push button station. This starter provides overload protection and low-voltage release or protection. When ordering push button specify whether low-voltage protection or low-voltage release is desired. The "start" and "stop" push button station may be omitted at $2.50 if another type of master switch is desired.

*j* (440 and 550 volts). Price includes a Class 11,500 AF enclosed magnetic auto-starter, "start" and "stop" push button station and "reset" push button. This starter provides overload protection and low-voltage release or protection. When ordering push buttons specify whether low-voltage protection or low-voltage release is desired. The "start" and "stop" push button station may be omitted at $5.00 if another type of master switch is desired.

*k* (2200 volts). Price includes a Class 11,800 magnetic starter and "start" and "stop" push button station. This starter provides overload protection. When ordering push button station specify whether low-voltage protection or low-voltage release is desired. "Start" and "stop" push button may be omitted at $2.50 if another type of master switch is desired.

*l* (220 volts). Price includes a Class 10,800 auto-transformer type starting panel with oil circuit-breakers for starting and running and separately mounted auto-transformer. This starter can be supplied for 3-phase circuits only, and provides overload and low voltage protection. Ammeter may be added at $43.00.

*m* (440, 550 and 2200 volts). Price includes a Class 10,700 type A enclosed auto-starter providing overload and low-voltage protection by automatically resetting relays.

*n* (220 volts). Price includes a Class 11,750 magnetic starter and "start" and "stop" push button station. This starter provides overload protection and low-voltage release or protection. When ordering push button specify whether low-voltage protection or low-voltage release is desired. The "start" and "stop" push button station may be omitted at $2.50 if another type of master switch is desired

*o* (440 and 550). Price includes a Class 11,550 enclosed magnetic starter and "start" and "stop" push button station. This starter provides overload protection and low-voltage release or protection. When ordering push button specify whether low-voltage protection or low-voltage release is desired. The "start" and "stop" push button may be omitted at $2.50 if another type of master switch is desired

*p* Special offset hub pulley.

Conduit boxes are regularly furnished for motor frames

- 200C to 600C inclusive.
- 400C to 700C inclusive.
- 600C to 800C inclusive.
- 700C to 800C inclusive. No allowance for omission.
- 900 if requested. No allowance for omission.
- 900 to 1000 if requested.

NOTE—Ball Bearings can be furnished on all motor frames in this table at an addition of 10 per cent to the bare motor price.

## TABLE 153

### Brine and Water Pumps

(Centrifugal Type)

| Model Pump | Horse Power Motor | Pumps | | Motors | | | Starters | | | Shipping Weight of Pump, Base and Polyphase Motor in. | Size of Pipe | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Price of bronze fitted or all iron pump with base and coupling | Price of all bronze pump with base and coupling | Price of induction 2 or 3 phase motors, 60 cycles, 110, 220 or 440 v., 1800 r.p.m. | Price of repulsion starter induction single phase motors, 60 cycles, 110-220 volts, 1800 r.p.m. | Price of d.c. constant speed motors, 115-230 volts, 1800 r.p.m. | Price of starters for a.c. 3 phase motors (for 2 phase, 4 wire add 40 per cent) | Price of starters for a.c. single phase motors | Price of starters for d.c. motors | | Size of Suction in. | Size of discharge in. |
| CC | ⅙ | $75.00 | $75.00 | ..... | $31.20 | ..... | ..... | $30.00 | ..... | 215 | 1 | ¾ |
| | ¼ | 75.00 | 75.00 | ..... | 37.80 | $52.00 | ..... | 30.00 | $49.00 | 215 | 1 | ¾ |
| | ½ | 75.00 | 75.00 | $49.30 | 53.60 | 70.40 | $30.00 | 30.00 | 49.00 | 215 | 1 | ¾ |
| AAB | ½ | 90.00 | 90.00 | 49.30 | 53.60 | 70.40 | 30.00 | 30.00 | 49.00 | 225 | 1 | ¾ |
| | ¾ | 90.00 | 90.00 | 58.00 | 72.50 | 84.80 | 30.00 | 30.00 | 49.00 | 235 | 1 | ¾ |
| LP | ¾ | 105.00 | 120.00 | 58.00 | 72.50 | 84.80 | 30.00 | 30.00 | 49.00 | 255 | 1½ | 1¼ |
| | 1 | 105.00 | 120.00 | 63.80 | 89.20 | 108.00 | 30.00 | 30.00 | 49.00 | 260 | 1½ | 1¼ |
| M-1 | ¾ | 115.00 | 130.00 | 58.00 | 72.50 | 84.80 | 30.00 | 30.00 | 49.00 | 240 | 1 | 1 |
| | 1 | 115.00 | 130.00 | 63.80 | 89.20 | 108.00 | 30.00 | 30.00 | 49.00 | 270 | 1 | 1 |
| | 1½ | 115.00 | 130.00 | 68.90 | 110.90 | 117.60 | 30.00 | 30.00 | 49.00 | 285 | 1 | 1 |
| L | ¾ | 120.00 | 135.00 | 58.00 | 72.50 | 84.80 | 30.00 | 30.00 | 49.00 | 255 | 1¼ | 1 |
| | 1 | 120.00 | 135.00 | 63.80 | 89.20 | 108.00 | 30.00 | 30.00 | 49.00 | 260 | 1¼ | 1 |
| | 1½ | 120.00 | 135.00 | 68.90 | 110.90 | 117.60 | 30.00 | 30.00 | 49.00 | 290 | 1¼ | 1 |
| MP | 1½ | 130.00 | 145.00 | 68.90 | 110.90 | 117.60 | 30.00 | 30.00 | 49.00 | 295 | 1½ | 1¼ |
| | 2 | 130.00 | 145.00 | 76.10 | 128.30 | 144.00 | 30.00 | 30.00 | 49.00 | 310 | 1½ | 1¼ |
| | 3 | 130.00 | 145.00 | 86.30 | 158.80 | 169.60 | 30.00 | 40.00 | 49.00 | 340 | 1½ | 1¼ |
| N | 1½ | 135.00 | 150.00 | 68.90 | 110.90 | 117.60 | 30.00 | 30.00 | 53.00 | 325 | 1½ | 1¼ |
| | 2 | 135.00 | 150.00 | 76.10 | 128.30 | 144.00 | 30.00 | 30.00 | 49.00 | 340 | 1½ | 1¼ |
| | 3 | 135.00 | 150.00 | 86.30 | 158.80 | 169.60 | 30.00 | 40.00 | 53.00 | 370 | 1½ | 1¼ |
| O | 2 | 140.00 | 160.00 | 76.10 | 128.30 | 144.00 | 30.00 | 30.00 | 49.00 | 345 | 2 | 1½ |
| | 3 | 140.00 | 160.00 | 86.30 | 158.80 | 169.60 | 30.00 | 40.00 | 53.00 | 375 | 2 | 1½ |
| | 5 | 140.00 | 160.00 | 105.80 | 208.00 | 252.80 | 30.00 | 40.00 | 53.00 | 400 | 2 | 1½ |

| | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| P | 2 | 145.00 | 165.00 | 76.10 | 128.30 | 144.00 | 30.00 | 30.00 | 49.00 | 350 | 2 | 1½ |
| | 3 | 145.00 | 165.00 | 86.30 | 158.80 | 169.60 | 30.00 | 40.00 | 53.00 | 380 | 2 | 1½ |
| | 5 | 145.00 | 165.00 | 105.80 | 208.00 | 252.80 | 30.00 | 40.00 | 53.00 | 405 | 2 | 1½ |
| S | 3 | 150.00 | 175.00 | 86.30 | 158.80 | 169.60 | 30.00 | 40.00 | 53.00 | 380 | 2½ | 2 |
| | 5 | 150.00 | 175.00 | 105.80 | 208.00 | 252.80 | 30.00 | 40.00 | 53.00 | 405 | 2½ | 2 |
| | 7½ | 150.00 | 175.00 | 130.50 | 284.00 | 307.20 | 40.00 | 50.00 | 64.00 | 465 | 2½ | 2 |
| LN | 3 | 155.00 | 180.00 | 86.30 | 158.80 | 169.60 | 30.00 | 40.00 | 53.00 | 390 | 2 | 1½ |
| | 5 | 155.00 | 180.00 | 105.80 | 208.00 | 252.80 | 30.00 | 40.00 | 53.00 | 415 | 2 | 1½ |
| | 7½ | 155.00 | 180.00 | 130.50 | 284.00 | 307.20 | 40.00 | 50.00 | 64.00 | 485 | 2 | 1½ |
| T | 3 | 170.00 | 195.00 | 86.30 | 158.80 | 169.60 | 30.00 | 40.00 | 53.00 | 390 | 2½ | 2 |
| | 5 | 170.00 | 195.00 | 105.80 | 208.00 | 252.80 | 30.00 | 40.00 | 53.00 | 415 | 2½ | 2 |
| | 7½ | 170.00 | 195.00 | 130.50 | 284.00 | 307.20 | 40.00 | 50.00 | 64.00 | 485 | 2½ | 2 |
| L-5 | 3 | 175.00 | 205.00 | 86.30 | 158.80 | 169.60 | 30.00 | 40.00 | 53.00 | 405 | 3 | 2½ |
| | 5 | 175.00 | 205.00 | 105.80 | 208.00 | 252.80 | 30.00 | 40.00 | 53.00 | 430 | 3 | 2½ |
| | 7½ | 175.00 | 205.00 | 130.50 | 284.00 | 307.20 | 40.00 | 50.00 | 64.00 | 500 | 3 | 2½ |
| LS | 7½ | 275.00 | 315.00 | 130.50 | 284.00 | 307.20 | 40.00 | 50.00 | 64.00 | 825 | 2½ | 2½ |
| | 10 | 275.00 | 315.00 | 155.00 | ..... | 352.00 | 40.00 | ..... | 64.00 | 935 | 2½ | 2½ |
| | 15 | 275.00 | 315.00 | 198.50 | ..... | 456.00 | 50.00 | ..... | ..... | 975 | 2½ | 2½ |
| | 20 | 275.00 | 315.00 | 243.50 | ..... | 544.00 | 60.00 | ..... | ..... | 1100 | 2½ | 2½ |
| LSS | 7½ | 285.00 | 325.00 | 130.50 | ..... | 307.20 | 40.00 | ..... | 64.00 | 885 | 2½ | 2½ |
| | 10 | 285.00 | 325.00 | 155.00 | ..... | 352.00 | 40.00 | ..... | 64.00 | 1000 | 2½ | 2½ |
| | 15 | 285.00 | 325.00 | 198.50 | ..... | 456.00 | 50.00 | ..... | ..... | 1050 | 2½ | 2½ |
| | 20 | 285.00 | 325.00 | 243.50 | ..... | 544.00 | 60.00 | ..... | ..... | 1160 | 2½ | 2½ |
| | 25 | 285.00 | 325.00 | 290.00 | ..... | 595.00 | 60.00 | ..... | ..... | 1300 | 2½ | 2½ |
| M-6 | 7½ | 335.00 | 380.00 | 130.50 | ..... | 307.20 | 40.00 | ..... | 64.00 | 825 | 4 | 3½ |
| | 10 | 335.00 | 380.00 | 155.00 | ..... | 352.00 | 40.00 | ..... | 64.00 | 935 | 4 | 3½ |
| | 15 | 335.00 | 380.00 | 198.50 | ..... | 456.00 | 50.00 | ..... | ..... | 975 | 4 | 3½ |
| | 20 | 335.00 | 380.00 | 243.50 | ..... | 544.00 | 60.00 | ..... | ..... | 1100 | 4 | 3½ |
| H-6 | 20 | 550.00 | 620.00 | 243.50 | ..... | 544.00 | 60.00 | ..... | ..... | 1250 | 3½ | 3 |
| | 25 | 550.00 | 620.00 | 290.00 | ..... | 595.00 | 60.00 | ..... | ..... | 1400 | 3½ | 3 |
| | 30 | 550.00 | 620.00 | 335.00 | ..... | 646.00 | 60.00 | ..... | ..... | 1400 | 3½ | 3 |

All prices are f.o.b. Iowa.

PUMPS.—Bronze fitted have Cast Iron casings with bronze impellers and packing glands with monel metal shafts.

All bronze pumps are of bronze construction throughout with monel metal shafts.

All iron pumps are of cast iron construction throughout with monel metal shafts.

All pumps have two outboard radial thrust ball bearings.

STARTERS.—Polyphase and single phase a.c. motor starters furnished are type enclosed in steel case for wall mounting with conduit knockouts; have thermal inverse time limit overload relays which are hand reset from outside of cabinet and low voltage protection. Prices include one "start" and "stop" push bottom station. Add 40 per cent to price for 2 phase 4 wire. When power company will not permit motor to be thrown directly across the line, then resistance type starters must be used, prices of which will be furnished on application. The J-1552 Starter may also be operated by float switch, pressure switch or thermostat.

Direct Current Starters listed are enclosed automatic starter with under voltage release and fused line switch.

Prices furnished on application for Direct Current Starters larger than 10 hp.

TABLE 154

CENTRIFUGAL PUMPS

Type "A" Side Suction Single-stage Centrifugal Pumps

| Size of Suction | Size of Discharge, Inches | Economic Capacity in Gallons per Minute | Approximate Weight, Pounds | Standard Pulley, Inches | | List Standard | List with Bronze Impeller | List for All Bronze Water End |
|---|---|---|---|---|---|---|---|---|
| | | | | Diameter | Face | | | |
| 2 | 1½ | 60 | 240 | 6 | 4 | $96 | $106 | $190 |
| 3 | 2 | 100 | 260 | 6 | 4 | 100 | 110 | 200 |
| 4 | 3 | 250 | 500 | 8 | 5 | 150 | 165 | 340 |
| 5 | 4 | 450 | 1200 | 10 | 6 | 350 | 390 | 780 |
| 6 | 5 | 750 | 1400 | 14 | 8 | 380 | 420 | 810 |
| 8 | 6 | 1000 | 1600 | 18 | 10 | 420 | 470 | 850 |
| 10 | 8 | 1800 | 2300 | 20 | 10 | 650 | 700 | 1350 |
| 12 | 10 | 3000 | 2700 | 22 | 12 | 720 | 780 | 1600 |
| 14 | 12 | 4250 | 3000 | 24 | 14 | 800 | 875 | 2000 |

Standard Pump fitted with Steel Shaft and Cast Iron Impeller. Electric includes Flexible Couplings, Extended Sub-base and mounting of motor when delivered to our factory. Belted includes standard pulley and outboard bearing.

TYPE "E" SIDE SUCTION VOLUTE CASING SINGLE-STAGE CENTRIFUGAL PUMPS

| Size of Suction | Size of Discharge, Inches | Economic Capacity in Gallons per Minute | Approximate Weight, Pounds | Standard Pulley, Inches | | List Standard | List with Bronze Impeller | List for All Bronze Water End |
|---|---|---|---|---|---|---|---|---|
| | | | | Diameter | Face | | | |
| 2 | 1½ | 60 | 240 | 6 | 4 | $96 | $106 | $190 |
| 3 | 2 | 100 | 260 | 6 | 4 | 100 | 110 | 200 |
| 4 | 3 | 250 | 500 | 8 | 5 | 150 | 165 | 340 |
| 5 | 4 | 450 | 700 | 10 | 6 | 180 | 200 | 375 |
| 6 | 5 | 750 | 1400 | 14 | 8 | 380 | 420 | 810 |
| 8 | 6 | 1000 | 1600 | 18 | 10 | 420 | 470 | 850 |
| 10 | 8 | 1800 | 1800 | 20 | 10 | 460 | 500 | 900 |

30 per cent discount from list price.

TABLE 155

Operating Labor Costs in Oil Engine Plants

[Power, 1925]

| Plant Number | Number of Units | Total Horse Power | Hours of Service per Day | Total Engine Hours per Year, All Units | Number of Attendants | Total Labor Cost, Dollars | Location |
|---|---|---|---|---|---|---|---|
| 1 | 2 | 150 | 24 | 7,552 | 2 | 3000 | North Dakota |
| 2 | 3 | 175 | 24 | 11,315 | 3 | 3900 | South Dakota |
| 3 | 2 | 250 | 24 | ..... | 3 | 4800 | Nebraska |
| 4 | 1 | 50 | 10 | 3,650 | 1 | 2160 | Nebraska |
| 5 | 1 | 37½ | 14 | 4,750 | 1 | 1620 | Iowa |
| 6 | 2 | 200 | 24 | 8,800 | 3 | 3440 | Iowa |
| 7 | 2 | 100 | 24 | 8,395 | 2 | 2400 | Nebraska |
| 8 | 3 | 237½ | 24 | 10,030 | 3 | 4320 | Nebraska |
| 9 | 2 | 300 | 24 | 8,459 | 3 | 4320 | Louisiana |
| 10 | 1 | 100 | 12 | 4,380 | 1 | 1900 | Ohio |
| 11 | 1 | 50 | 24 | 8,010 | 1* | 1200 | California |
| 13 | 2 | 300 | 24 | 8,760 | 3 | 4440 | Michigan |
| 14 | 2 | 250 | 24 | 6,200 | 3 | 2270 | California |
| 15 | 2 | 150 | 8 | 2,920 | 1 | 1200 | Oklahoma |
| 16 | 1 | 150 | 24 | 6,570 | 2 | 1740 | Arkansas |
| 17 | 3 | 250 | 24 | 14,620 | 3 | 4800 | Texas |
| 18 | 1 | 75 | ..... | ..... | 2 | 3000 | Illinois |
| 19 | 4 | 900 | 24 | ..... | 4 | 5760 | Texas |
| 20 | 3 | 350 | 24 | ..... | 2 | 2820 | Texas |
| 21 | 1 | 100 | 8 | 2,920 | 1 | 900 | Kentucky |
| 22 | 2 | 100 | 24 | 10,220 | 2 | 2400 | Oklahoma |
| 23 | 2 | 75 | 10 | 4,350 | 2 | 2500 | Maine |
| 24 | 3 | 600 | 24 | 5,760 | 3 | 2310 | Tennessee |
| 25 | 2 | 300 | 24 | 12,000 | 2 | 2010 | Mississippi |
| 26 | 2 | 150 | 24 | 5,923 | 2 | 3206 | Texas |
| 27 | 3 | 300 | 24 | 12,720 | 2 | 2300 | Mississippi |

* One man on each shift operates ice plant.

TABLE 156

AN ANALYSIS OF REPAIR COSTS IN THE OIL ENGINE PLANTS

| Plant Number | Number of Units | Total Horse Power | Years of Service | Total Cost of Engine Repairs, Dollars | Extra Labor Cost, Dollars | Repairs per Unit Produced, Dollars |
|---|---|---|---|---|---|---|
| 1 | 2 | 150 | 1½ | | | |
| 2 | 3 | 175 | 8 | 2506.00 | ..... | 0.00024 |
| 3 | 2 | 250 | 1 | | | |
| 4 | 1 | 50 | 2 | 0.84 | | |
| 5 | 1 | 37½ | 3 | | | |
| 6 | 2 | 200 | 1 | | | |
| 7 | 2 | 100 | 6 | 1007.68 | ..... | 0.00016 |
| 8 | 3 | 237½ | 5 | 980.88 | 40.00 | |
| 9 | 2 | 300 | 4 | 150.00 | | |
| 10 | 1 | 100 | 1½ | 65.00 | 50.00 | |
| 11 | 1 | 50 | 10 | 265.00 | | |
| 12 | 1 | 100 | 4 | | | |
| 13 | 2 | 300 | 4 | 785.00 | ..... | 0.0005 |
| 14 | 2 | 250 | 8 | 368.00 | | |
| 15 | 2 | 150 | 4 | 51.51 | 50.00 | |
| 16 | 1 | 150 | 1½ | | | |
| 17 | 3 | 250 | 10 | 6293.00* | ..... | 0.0012 |
| 18 | 1 | 75 | 3 | 249.03 | | |
| 19 | 4 | 900 | 2 | 605.15 | ..... | 0.00024 |
| 20 | 3 | 350 | 1 | | | |
| 21 | 1 | 100 | 1 | | | |
| 22 | 2 | 100 | 6 | 750.00 | | |
| 23 | 2 | 75 | 6 | 1554.12 | | |
| 24 | 3 | 600 | 1 | | | |
| 25 | 2 | 300 | 1 | 21.65 | | |
| 26 | 2 | 150 | 4 | 527.67 | | |
| 27 | 3 | 300 | 4 | 308.46 | ..... | 0.00027 |
| 28 | 2 | 400 | 2 | 414.90 | 523.98 | |
| 29 | 1 | 100 | 3 | 54.04 | | |

* Crankshaft.

TABLE 157

Operating Costs of the Oil-engine Driven Electric Plants

| Plant Number | Load Factor, Per Cent | | Kw.-Hr. Produced per Year | Operating Cost, Dollars | |
|---|---|---|---|---|---|
| | Running | Yearly | | Total | Unit |
| 1 | 37½ | 15 | 113,605 | 4,929 | 0.0434 |
| 2 | 42 | 18 | 165,050 | 6,787 | .0411 |
| 6 | 82 | 41 | 430,000 | 7,320 | .0170 |
| 7 | 48 | 23 | 120,000 | 4,080 | .0340 |
| 13 | 52 | 36 | 403,000 | 8,497 | .0210 |
| 17 | 58 | 37 | 481,800 | 10,309 | .0214 |
| 19 | .... | 36 | 1,271,400 | 15,224 | .0120 |
| 24 | 57 | 21 | 650,000 | 10,236 | .0157 |
| 27 | 38 | 23 | 365,000 | 5,648 | .0155 |
| | | | | Average.... | 0.246 |

| Plant Number | Total Investment, Dollars | Fixed Charge Cost, Dollars | | Unit Fuel and Lubricating Oil Cost, Dollars | Unit Total Cost with Fixed Charges, Dollars |
|---|---|---|---|---|---|
| | | Total | Unit | | |
| 1 | 19,800 | 1980 | 0.0174 | 0.016 | 0.0608 |
| 2 | 36,300 | 3630 | .0220 | .0147 | .0631 |
| 6 | 30,000 | 3000 | .0070 | .0066 | .0240 |
| 7 | 18,200 | 1820 | .0152 | .0140 | .0490 |
| 13 | 41,000 | 4100 | .0102 | .0094 | .0312 |
| 17 | 34,400 | 3440 | .0071 | .0079 | .0285 |
| 19 | 74,400 | 7440 | .0058 | .0074 | .0178 |
| 24 | 92,000 | 9200 | .0141 | .008 | .0298 |
| 27 | 54,000 | 5400 | .0148 | .009 | .0303 |
| | | Average.... | 0.0124 | 0.0103 | |

TABLE 158

OPERATING COSTS OF THE OIL ENGINE PUMPING PLANTS COMPARED WITH STEAM- AND MOTOR-DRIVEN PLANTS

| Plant | Number of Pumps | Gallons Pumped, Millions | Static Head | Operating Cost Except Fuel and Lubricating Oil, Dollars | Cost per Million Gallon, Cent | Fuel and Lubricating Oil, Dollars | Fuel and Lubricating Oil per Million Gallons, Cents | Total Operating Costs per Million Gallons, Cents | Fuel and Lubricating Oil Cost Corrected for 150-Ft. Head, Cents | Total Operating Costs on Basis of 150-Ft. Head, Cents | Price of Fuel Oil Coal Current, Cents per Gallon |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 5 | 2 | 142 | 150 | 1,805 | 12.72 | 1,505 | 10.60 | 23.32 | 11.12 | 23.84 | 11.9 |
| 10 | 1 | 163.3 | 322 | 2,115 | 12.92 | 1,532 | 9.38 | 22.30 | 4.37 | 17.29 | 6.47 |
| 15 | 2 | 87.6 | 402 | 1,260 | 14.38 | 619 | 7.07 | 21.45 | 2.64 | 17.02 | 4.0 |
| 16 | 1 | 222.0 | 402 | 2,040 | 9.20 | 2,568 | 11.55 | 20.75 | 4.32 | 13.52 | 6.02 |
| 26 | 2 | 262.1 | 200 | 4,046 | 15.41 | 1,457 | 5.56 | 20.97 | 4.17 | 19.58 | 3.57 |
| **Steam Pumping Plant Costs** | | | | | | | | | | | |
| | | | | | | | | | | | Dollars per Ton |
| Marinette | 4 | 617.0 | 150 | 4,142 | 6.70 | 15,806 | 25.60 | 32.30 | 25.60 | 32.30 | 6.72 |
| La Crosse | 8 | 1213.0 | 240 | 18,780 | 15.45 | 19,419 | 16.00 | 31.45 | 10.00 | 25.45 | 6.89 |
| Berlin | 2 | 77.4 | 135 | 4,066 | 52.70 | 2,093 | 27.10 | 79.80 | 30.50 | 83.20 | 7.00 |
| Washburn | 2 | 155.8 | 250 | 2,840 | 18.25 | 6,186 | 39.75 | 58.00 | 23.80 | 42.05 | 9.33 |
| Janesville | 4 | 589.0 | 175 | 6,760 | 11.45 | 7,405 | 12.55 | 24.00 | 10.75 | 22.20 | 6.04 |
| Kenosha | 5 | 1670.0 | 162 | 17,491 | 10.45 | 19,816 | 11.85 | 22.30 | 10.97 | 21.42 | 7.60 |
| South Milwaukee | 6 | 317.0 | 210 | 11,536 | 36.50 | 11,847 | 37.40 | 73.90 | 26.70 | 63.20 | 7.60 |
| **Motor-driven Pumping Plant Costs** | | | | | | | | | | | |
| | | | | | | | Current Costs | | | | Cents per Kw.-Hr. |
| Menominee | 2 | 170.0 | 240 | 3,361 | 19.80 | 5,633 | 33.10 | 52.90 | 20.70 | 40.50 | 1.25 |
| Delavan | 3 | 244.3 | 125 | 3,560 | 145.80 | 1,378 | 56.60 | 202.40 | 68.00 | 213.80 | 3.0 |
| Columbus | 2 | 466.5 | 135 | 767 | 16.50 | 1,573 | 34.80 | 51.30 | 38.70 | 55.20 | 3.5 |
| Castleton | 1 | 146.7 | 223 | 1,374 | 93.50 | 1,533 | 104.20 | 197.70 | 70.40 | 163.90 | 4.5 |
| Neenah | 3 | 177.0 | 110 | 3,642 | 20.60 | 8,246 | 46.50 | 67.10 | 39.40 | 60.00 | |
| Wisconsin Rapids | 4 | 214.0 | 195 | 2,619 | 12.20 | 5,034 | 23.50 | 35.70 | 18.05 | 30.25 | 2.44 |
| Chippewa Falls | 4 | 371.6 | 185 | 2,471 | 6.65 | 6,475 | 17.40 | 24.05 | 14.10 | 20.75 | 1.44 |
| Lake Geneva | 4 | 608.7 | 186 | 3,531 | 58.00 | 4,132 | 67.80 | 125.80 | 54.70 | 102.70 | 3.0 |
| Stoughton | 3 | 150.4 | 185 | 2,904 | 19.35 | 3,190 | 21.20 | 40.55 | 17.20 | 36.55 | 2.0 |
| Waukesha | 8 | 536.6 | 150 | 7,133 | 13.28 | 18,494 | 34.50 | 34.78 | 34.50 | 47.78 | 2.39 |

NOTE.—Motor and steam plant costs from Railroad Commission, State of Wisconsin.

TABLE 159

DEPRECIATION OF ICE PLANTS—G. E. WELLS, A.S.R.E., 1920

| | Per cent |
|---|---|
| Boilers | 10 |
| Pipe work | 5 |
| Brick building | 2 to 3 |
| Electric machinery | 7 |
| General machinery | 7 |
| Automatic equipment | 20 |
| Delivery equipment | 10 |
| Country ice houses | 20 |
| Office furniture and fixtures | 10 |
| Tools and implements | 10 |

TABLE 160

ESTIMATED YEARLY COST OF ELECTRIC PLANT OPERATION
[Power, 1926]

Output 30,000 tons annually. Based on actual operation, June to October, 1926

| | | |
|---|---|---|
| Investment: | | |
| All electric equipment including installation and wiring cost | | $25,000.00 |
| Fixed charges: | | |
| Interest, per cent | 6 | |
| Depreciation, per cent | 4 | |
| Insurance and taxes, per cent | 2 | |
| Total fixed charges | 12 | $3,000.00 |
| Electricity: | | |
| 30,000 × 60 kw.-hr. per ton = 1,800,000 kw.-hr., at 1.14c | | 20,500.00 |
| Fuel for heating: | | |
| 20 tons coal at $6 | | 120.00 |
| Labor: | | |
| 1 Chief engineer at $45.00 | $2340.00 | |
| 2 Night engineers at $38.00 | 3960.00 | |
| 1 Electrician at 30.00 | 1560.00 | |
| 3 Oilers and ice pullers at $25.00 | 3900.00 | |
| Total | | 11,760.00 |
| Supplies and repairs: | | |
| 25c. per ton × 30,000 | | 7,500.00 |
| Total | | $42,880.00 |
| Cost per ton | | 1.43 |

TABLE 161

ESTIMATED YEARLY COST OF OIL ENGINE PLANT OPERATION AND INCLUDING FIXED CHARGES

Output 30,000 Tons Annually

| | | |
|---|---|---|
| Investment: | | |
| Two 200-hp. Diesel engines installed | | $26,000.00 |
| Two 200-hp. Diesel engines driving 140-kw. generators | | 33,000.00 |
| 200 hp. of electric motors, wiring and installation | | 7,000.00 |
| | | $66,000.00 |
| Fixed charges: | | |
| Interest, per cent | 6 | |
| Depreciation, per cent | 8 | |
| Insurance and taxes, per cent | 2 | |
| Total | 16 | $10,580.00 |
| Fuel: | | |
| 6 gal. per ton × 30,000 = 180,000 gal. at 6½ c | | 11,700.00 |
| Lubricating oil: | | |
| 1-10 gal. per ton × 30,000 = 3,000 gal. at 55c | | 1,650.00 |
| Labor: | | |
| 1 Chief engineer at $45.00 | $2340.00 | |
| 2 Watch engineers at $38.00 | 3960.00 | |
| 1 Electrician at $30.00 | 1560.00 | |
| 4 Oilers and ice pullers at $25.00 | 5200.00 | |
| Total | | $13,060.00 |
| Supplies and repairs: | | |
| 30 c. per ton × 30,000 | | 9,000.00 |
| Total | | $45,990.00 |
| Cost per ton | | 1.53 |

TABLE 162

Operation Costs With and Without Ice Storage

| | 200-Ton Plant | | 10,000-Ton Storage and 125-Ton Plant | |
|---|---|---|---|---|
| | Cost per ton | Total cost | Cost per ton | Total cost |
| General expense, including insurance, taxes, depreciation and interest on investment (14 per cent) | $1.68 | $4,200 | $1.55 | $38,780 |
| Plant labor | 0.10 | 2,500 | 0.14 | 3,500 |
| Tank-room labor | 0.10 | 2,500 | 0.14 | 3,500 |
| Day storage-room labor | 0.06 | 1,500 | 0.10 | 2,500 |
| Power | 0.45 | 11,250 | 0.47 | 11,750 |
| Ammonia | 0.05 | 1,250 | 0.05 | 1,250 |
| Water and other supplies | 0.05 | 1,250 | 0.05 | 1,250 |
| Maintenance | 0.15 | 3,750 | 0.15 | 3,750 |
| Salary of portion of crew during shutdown period | 0.15 | 3,850* | 0.088 | 2,200† |
| Winter storage-room labor 10,000 tons | ..... | ..... | 0.40 | 4,000 |
| Winter storage-room refrigeration 10,000 tons | ..... | ..... | 0.30 | 3,000 |
| Totals | ..... | $69,850 | ..... | $75,480 |
| Average cost per ton | $2.79 | ..... | $3.02 | |

Saving in favor of large daily capacity, $0.23 per ton.

* On basis of 5 months' operation; shutdown 7 months at $550 per month.
† On basis of 8 months' operation; shutdown 4 months at $550 per month.

## CHAPTER XXI

## SPECIFICATIONS

The object of specifications is to set down concisely, yet clearly and completely, all the requirements of a purchase. Either of two methods may be pursued. First the purchaser can state exactly what is desired in the apparatus or the machinery, and material entering into the construction and the types and the construction of the accessories and ask for no guarantee. Second, the purchaser can state the capacity required, or the results desired, and leave to the manufacturer the obligation to work out the details of the design and then specify a guarantee of performance. In the second case the reputation of the manufacturer is an important factor, as frequently the difficulty of making tests or of proving that the guarantee is lived up to, makes it of no value. As a rule it is much more satisfactory to specify standard equipment, for in such cases the price is usually considerably less than would be the case where new patterns and possibly special machinery must be provided in order to manufacture the special product.

**Compressors.**—The purchaser must needs make a choice of the type of compressor he desires. In America this choice is confined to ammonia and carbon dioxide as refrigerants. Although carbon dioxide is usually stated as a cold condensing water refrigerant, it is used to a very large extent on the merchant marine and even (to a slight extent) for stationary work in the southern part of the United States and Cuba. Notwithstanding that the power requirements are greater for carbon dioxide than for ammonia and that these increase with the temperature of the "liquid" from the condenser, yet the excess power for carbon dioxide over that for ammonia is only nominal. The real decision as to the kind of refrigerant should be made because of other factors such as safety, compactness, kind of load, etc.

The type of compressor and the kind of drive should be specified. The preferred types are: for 10 tons or less, the automatic; for 75 tons and less, the twin cylinder, vertical, single-acting machine; and for over 75 tons, either the horizontal double-acting type or the vertical single-acting machine with the open frame. There are decided advantages in the use of the enclosed compressor which in Great Britain, in particular,

are evidenced by the use of three or four cylinders on the same shaft with capacities of 200 tons and over. [1]

The rotative speed of the compressor should be specified. Speeds over 100 r.p.m. require careful machine design as regards the type of valve and the valve opening, stresses in the frame, reciprocating parts and bearings, and very particular attention to the lubrication. The old packing house design, steam engine driven, has become obsolete and the synchronous motor drive (where electric power is available) and the uniflow or poppet valve medium speed steam engine and oil engine driven compressor are replacing it. Experience has shown that the essential point to be considered in any case is the perfection of the lubrication which must be positive and sufficient in amount.[2] The use of stage compression should be specified for evaporating temperatures of 0 deg. F. and lower, especially where the condenser pressure is 150 lb. gage and over. Multiple effect compression has certain limited advantages in special cases and it will be worth consideration when two evaporating pressures are carried. Multiple effect compression can be used to advantage in carbonic compression for cooling the liquid refrigerant, and particularly when the water for the condenser is 70 deg. F. and higher.

Details on the design of the piping should be specified in full. The gas flow into and out of the compressor should be as direct as possible. Metallic packing causes much less friction than does fibre packing or leather cups, will not wear the rod as much and can be made gas-tight. By-pass valves and connections should be of ample size and convenient of operation. Means should be provided for suction from the atmosphere, when testing out under air pressure, and discharge to the atmosphere, when pumping a vacuum, without having to remove fittings or flange connections. The oil and scale trap should be of ample size and should be designed for convenience as to the removal of oil and scale. For carbonic refrigeration it is well to specify that the scale trap, oil separator, etc., should be welded. Mention should be made of the desired size of the suction and condenser pressure gages, and the size and the location of the safety pop valve, whether discharging to the atmosphere or to the low-pressure side of the system.

**Condensers.**—Ammonia condensers should be erected with purge and pump-out connections. The amount of water available, and its temperature, should be mentioned, and the kind of condenser should be specified as well as the amount of the condensing surface, which in all cases should be liberal.[3] The modern condensers for ammonia are the

[1] See Chapter II.

[2] See construction details in Chapter II.

[3] See Chapter VI.

common atmospheric, the drip type, the double pipe, and the shell and tube. Compactness and moderate efficiency have brought the shell and tube condenser into popularity, especially in the larger sizes. Of the pipe condensers the common and the drip type are the popular atmospheric condensers in the order named, and the double pipe is used where the water is reasonably clear and particularly for automatic and other small installations. In carbonic work the welded double pipe and the modified submerged coil type are preferred. The number of pipes per stand should be mentioned, although it is doubtful whether more than twelve pipes are economical. The number and the spacing of the stands, the piping arrangement of the headers and the liquid drain, the kind of fittings, the water distributor and valves should be specified. If the spacing of the pipes for the atmospheric condenser is $4\frac{1}{2}$ in. or over, splash strips should be provided. As some manufacturers make use of full weight steel pipe, whereas others use extra heavy steel or wrought iron, the kind of pipe should be mentioned.

*The Liquid Receiver* should be specified by the diameter and length, and the thickness of the shell and of the heads should be given. The liquid receiver should be tested before shipment with a 500 lb. air pressure under water. The volume of the receiver should bear a relationship to the amount of the liquid in the system (see Chapter X), and in the smaller sizes the ability to store the entire amount of the charge in the receiver is very useful. The gage glasses should be protected in an approved manner, and a safety valve to prevent an excess pressure in the receiver should be required.[4]

**Fittings.**—Fittings may be screwed or flanged; of malleable iron, semi-steel or drop forged. The extensive use of welding has stimulated the use of steel flanges particularly, as well as steel fittings and valves because of the ease in welding, especially in the case of carbonic refrigeration. As a rule the screw fitting is confined to the smaller pipe sizes, whereas the larger sized pipe uses the oval, the square, or the round flange, depending on the size. The accepted standard flange has the tongue and groove, using a lead, rubber or asbestos gasket. In carbonic refrigeration it has been found that the only tight flange is that made with the end of the pipe and a hard gasket. Care should be taken in the specifications so as to include the kind of fittings desired.

Practically all ammonia valves are designed so as to permit packing of the valve stem while the valve is under pressure. Either the soft or the hard seat valve may be specified, and both have relative advantages. The soft seat valve will be tight until the lead is squeezed out and

[4] See details on liquid receivers in Chapter IV.

reseating becomes necessary. The scale trap should be mentioned, giving details for cleaning.

**Piping.**—Steel pipe now is used almost exclusively in ammonia refrigeration, although wrought iron is still specified. There is no standard weight of pipe in the high-pressure side, as a number of manufacturers use full weight pipe. Likewise both full weight and extra heavy pipe is specified for the direct expansion piping in ice tanks. It is usual to specify butt welded pipe in 2-in. sizes and under, and lap welded pipe in sizes over 2 in.

In ammonia direct expansion piping fittings are dispensed with as much as possible. The "hair pin" coil is now used in many cases where continuous welding is not used. Detail drawing should be made of all piping, both on the high and the low-pressure side.

Ice making tanks should be given the exact dimensions desired, including the thickness of the plate, the bulkheads, partitions, scarfing of the seams, reinforcing, the boxes and the drains. If a brine cooler is used, the diameter, length and the amount of surface desired should be given. The tubes are frequently 2-in. o.d. of No. 12 charcoal iron and are expanded into the tube sheet, which is approximately $\frac{1}{2}$ in. thick. A single pass is used for ice tanks and 4, or even as many as 8, passes for general brine cooling. A pump out, oil drain and liquid gage connection should be provided.

The size and the number of cans, the amount of agitation of the brine and the details of the air agitation in the cans should be specified (Chapter XIII). The method of water softening, the use of cooling towers or of sprays, the kind of crane for the cans and the number of cans to be lifted at a time, the use of baskets, the manner of thawing, etc., should be specified and not left to the manufacturer.

**Brine.**—Calcium chloride brine is specified for most large work not using brine sprays. Such brine is usually best when it is free from magnesium, and it should be of such a concentration as will be safe from freezing at the lowest operating temperature.[5]

The following tables give some idea of the detailed weights and sizes of standard equipment.

[5] See Chapter VIII.

TABLE 163

LOW-PRESSURE FREEZING SYSTEM APPARATUS

| Tons of Ice | Number of Cans | Air Blowers | | | | | | | | Core Pumping Set | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Number of blowers | Size of pipe, inches | C.f.m. | R.p.m. | Horse power motor required | Size of blower pulley, inches | Belted, weight, pounds, | Maximum r.p.m. motor | Number | Size of pump | Belted, weight, pounds | Motor-driven, weight, pounds | Horse power motor required |
| 1 | 16 | 1 | $2\frac{1}{2}$ | 8 | 520 | .75 | 7×1.5 | 90 | 1200 | 1 | $1\frac{1}{4}$ | 200 | 260 | 2 |
| 2 | 30 | 1 | $2\frac{1}{2}$ | 15 | 635 | .75 | 7×1.5 | 90 | 1200 | 1 | $1\frac{1}{4}$ | 200 | 260 | 2 |
| 3 | 42 | 1 | $2\frac{1}{2}$ | 21 | 600 | 1.5 | 8×2 | 200 | 1200 | 1 | $1\frac{1}{4}$ | 200 | 260 | 2 |
| 4 | 56 | 1 | $2\frac{1}{2}$ | 28 | 670 | 1.5 | 8×2 | 200 | 1200 | 1 | $1\frac{1}{4}$ | 200 | 260 | 2 |
| 5 | 70 | 1 | $2\frac{1}{2}$ | 35 | 740 | 2.0 | 8×2 | 200 | 1800 | 1 | $1\frac{1}{4}$ | 200 | 260 | 2 |
| 6 | 84 | 1 | $2\frac{1}{2}$ | 42 | 810 | 2.0 | 8×2 | 200 | 1800 | 1 | $1\frac{1}{4}$ | 200 | 260 | 2 |
| 8 | 112 | 1 | $2\frac{1}{2}$ | 56 | 700 | 3.0 | 10×2.5 | 300 | 1800 | 1 | $1\frac{1}{4}$ | 200 | 260 | 2 |
| 10 | 144 | 1 | $2\frac{1}{2}$ | 72 | 800 | 3.0 | 10×2.5 | 300 | 1800 | 1 | $1\frac{1}{4}$ | 200 | 260 | 2 |
| 12 | 168 | 1 | 3 | 84 | 505 | 5.0 | 12×3.0 | 400 | 1800 | 1 | $1\frac{1}{4}$ | 200 | 260 | 2 |
| 15 | 210 | 1 | 3 | 105 | 550 | 5.0 | 12×3.0 | 400 | 1200 | 1 | $1\frac{1}{4}$ | 200 | 260 | 2 |
| 20 | 280 | 1 | 3 | 140 | 660 | 5.0 | 12×3.0 | 400 | 1200 | 1 | $1\frac{1}{4}$ | 200 | 260 | 2 |
| 25 | 360 | 1 | 3 | 180 | 790 | 5.0 | 12×3.0 | 400 | 1200 | 1 | $1\frac{1}{4}$ | 200 | 260 | 2 |
| 30 | 420 | 1 | 4 | 210 | 460 | 7.5 | 16×3.0 | 730 | 1200 | 1 | $1\frac{1}{4}$ | 200 | 260 | 2 |
| 35 | 484 | 1 | 4 | 242 | 505 | 7.5 | 16×3.0 | 730 | 1200 | 1 | $1\frac{1}{4}$ | 200 | 260 | 2 |
| 40 | 560 | 1 | 4 | 280 | 565 | 7.5 | 16×3.0 | 730 | 1200 | 1 | $1\frac{1}{4}$ | 200 | 260 | 2 |
| 50 | 720 | 1 | 4 | 360 | 680 | 7.5 | 16×3.0 | 730 | 1200 | 1 | $1\frac{1}{4}$ | 200 | 260 | 2 |
| 60 | 840 | 2 | 4 | 420 | 460 | 7.5 | 16×3.0 | 1460 | 1200 | 1 | $1\frac{1}{4}$ | 200 | 260 | 2 |
| 80 | 1120 | 2 | 4 | 560 | 565 | 7.5 | 16×3.0 | 1460 | 1200 | 1 | $1\frac{1}{4}$ | 200 | 260 | 2 |
| 100 | 1440 | 2 | 4 | 720 | 680 | 7.5 | 16×3.0 | 1460 | 1200 | 1 | $1\frac{1}{4}$ | 200 | 260 | 2 |
| 120 | 1680 | 2 | 6 | 840 | 500 | 10.0 | 20×4.0 | 3000 | 1200 | 1 | $1\frac{1}{4}$ | 200 | 260 | 2 |
| 150 | 2160 | 2 | 6 | 1080 | 550 | 15.0 | 20×4.0 | 3000 | 1200 | 1 | $1\frac{1}{4}$ | 200 | 260 | 2 |
| 200 | 2880 | 3 | 6 | 1440 | 550 | 15.0 | 20×4.0 | 4500 | 1200 | 2 | $1\frac{1}{4}$ | 400 | 520 | 2 |

Core Pumping Set consists of centrifugal pump, hydraulic ejector and water tank and connections.
Blowers operate at 3 lb. gage pr. Use 1800 r.p.m. motors for motor-driven core pumping sets.

## TABLE 164

### 6-ton Distilled Water Ice Making Plant—Flooded (300-Lb. Cans)

| Items | Weight, Pounds | List Price |
|---|---|---|
| 1 Refrigerating machine, 7½×10½×6 single column | 11,000 | 1550 |
| Painting machine at the factory | ..... | 27 |
| 1 Double pipe ammonia condenser, 8 pipes high, 30 ft. long, 1½×2; black, full weight, wrought iron pipe | 1,950 | 155 |
| 1 Liquid receiver, welded, 12 in. dia by 12 ft. long | 600 | 72 |
| 1 Oil separator, welded, 12 in. dia. by 3 ft. long | 160 | 25 |
| Ammonia pipe connections, extra heavy discharge, full weight suction | 1,000 | 191 |
| 1 Freezing tank, 23 ft. by 11 ft. 4 in by 4 ft., ¼ in. steel | 5,900 | 300 |
| 1080 ft. 1¼ in. full weight black steel tank coils with stands and headers | 4,750 | 272 |
| 1 Accumulator, welded, 16 in. dia. by 4 ft. 9 in. long | 230 | 29 |
| 72 Oak covers and frames with can guides | 2,880 | 97 |
| 72 Freezing cans, 300 lb., 11½×22½×45 | 5,040 | 364 |
| 1 Propeller agitator, 15 in. diameter | 410 | 38 |
| Bulkhead and partitions for tank | 400 | 18 |
| 1 Crane and hand hoist, 300 lb. single can | 1,220 | 94 |
| 1 Can dump 300 lb. single iron type | 450 | 40 |
| 1 Can filler and 20 ft. 5-ply hose | 40 | 31 |
| 1 Exhaust steam oil separator, welded, 16 in. by 4 ft. long | 210 | 36 |
| 1 Shell steam condenser, 9 ft. 9 in. long, No. 10, iron, galvanized and trough | 580 | 61 |
| 1 Distilled water pump, steam driven, 4 gal. per min | 100 | 42 |
| 1 Float tank, 15 in. dia. by 24 in. long and float control | 115 | 22 |
| 1 Reboiler 7 in.×16 in.×18 in. and steam coil, painted | 380 | 50 |
| 1 Automatic regulator, ¾ in. connection | 75 | 14 |
| 1 Double pipe distilled water cooler, 6 pipes high, 10 ft. long, 1¼×2 in. full weight galvanized | 480 | 50 |
| 2 Distilled water filters, 24 in. dia. by 5 ft. long with charcoal to charge | 1,230 | 108 |
| 1 Distilled water storage tank, 7 ft.×4 ft.×3 ft. 6 in. $\frac{3}{16}$ in. iron, with 68 ft. extra heavy 2 in. pipe | 1,560 | 128 |
| Distilled water pipe connections, galvanized | 300 | 146 |
| 1 Horizontal return tubular boiler complete, 54 in. dia.×14 ft. long, 60 hp. | 15,665 | 993 |
| 1 Boiler feed pump, steam driven, 9 gal. per min | 180 | 63 |
| 1 Feed water heater, 60 hp. vertical closed type, brass tubes | 550 | 103 |
| Steam and exhaust pipe connections | 600 | 218 |
| 4½ Tons of salt | 9,000 | 58 |
| 300 Lb. ammonia | 300 | 41 |
| Engineer | ..... | 585 |
| Usual labor | ..... | 585 |
| Union labor | ..... | 1170 |
| 5¼ Tons calcium | 10,500 | 133 |

TABLE 165

1¼-In. and 2-In. Double Pipe Ammonia Condensers, 10 and 20 Ft. Long, Specifications, Weights and Prices

| Pipes, High, Inches | Effective Surface, Square feet | Height Over Top Pipe, Inches | Height Over All, Inches | Full Weight of Pipe, Pounds | Extra Heavy Pipe, Pounds | Add for Soldered Joints |
|---|---|---|---|---|---|---|
| | | | 10-Ft. Coils | | | |
| 6 | 21 | 32 | 47 | 750 | 840 | 17 |
| 8 | 28 | 40 | 55 | 1000 | 1120 | 22 |
| 10 | 35 | 48 | 63 | 1250 | 1400 | 28 |
| 12 | 42 | 56 | 71 | 1400 | 1625 | 33 |
| 14 | 49 | 64 | 79 | 1700 | 1900 | 39 |
| | | | 20-Ft. Coils | | | |
| 4 | 31.5 | 24 | 39 | 720 | 900 | 11 |
| 6 | 47 | 32 | 47 | 1080 | 1320 | 17 |
| 8 | 62.5 | 40 | 55 | 1440 | 1760 | 22 |
| 10 | 78 | 48 | 63 | 1800 | 2200 | 28 |
| 12 | 94 | 56 | 71 | 2100 | 2620 | 33 |

Gas, liquid and purge valves and 2 pr. flanges for water connections included. No headers nor water valves included.

## TABLE 166

### Condenser Pans

| Ammonia Condenser Pans | | | Flask Steam Condenser Pans | | | Steam Condenser and Distilled Water Cooler Pans | | |
|---|---|---|---|---|---|---|---|---|
| Number of coils | Size of pan | Weight, pounds | Number of flasks | Size of pan | Weight, pounds | Number of coils | Size of pan | Weight, pounds |
| | Ft. Ft. In. | | | Ft. Ft. In. | | | Ft. Ft. In. | |
| 1 | 14× 4×6 | 600 | 1 | 12× 4 ×6 | 520 | 1 | 12× 4×6 | 520 |
| 1 | 24× 4×6 | 1000 | 1 | 15× 4 ×6 | 640 | 1 | 22× 4×6 | 920 |
| 2 | 24× 6×6 | 1400 | 1 | 18× 4 ×6 | 760 | 2 | 22× 6×6 | 1300 |
| 3 | 24× 7×6 | 1600 | 1 | 22× 4 ×6 | 920 | 3 | 22× 8×6 | 1925 |
| 4 | 24× 9×6 | 2000 | 2 | 12× 6 6×6 | 780 | 4 | 22×10×6 | 2100 |
| 5 | 24×10×6 | 2200 | 2 | 15× 6 6×6 | 960 | 5 | 22×12×6 | 2550 |
| 6 | 24×12×6 | 2700 | 2 | 18× 6 6×6 | 1200 | 6 | 22×14×6 | 3000 |
| 7 | 24×13×6 | 3000 | 2 | 22× 6 6×6 | 1400 | 7 | 22×16×6 | 3400 |
| 8 | 24×15×6 | 3400 | 3 | 15× 9 ×6 | 1300 | 8 | 22×18×6 | 3800 |
| 9 | 24×16×6 | 3800 | 3 | 18× 9 ×6 | 1600 | 9 | 22×20×6 | 4200 |
| 10 | 24×18×6 | 4200 | 3 | 22× 9 ×6 | 1900 | 10 | 22×22×6 | 4600 |
| 11 | 24×19×6 | 4400 | 4 | 22×11 6×6 | 2450 | 11 | 22×24×6 | 5000 |
| 12 | 24×21×6 | 4800 | 5 | 22×14 ×6 | 3000 | 12 | 22×26×6 | 5400 |
| 13 | 24×22×6 | 5000 | 6 | 22×16 6×6 | 3500 | 13 | 22×28×6 | 5800 |
| 14 | 24×24×6 | 5400 | 7 | 22×19 ×6 | 4000 | 14 | 22×30×6 | 6200 |
| 15 | 24×25×6 | 5600 | 9 | 22×24 ×6 | 5000 | 15 | 22×32×6 | 6600 |
| 16 | 24×27×6 | 6000 | 11 | 22×29 ×6 | 5900 | 16 | 22×34×6 | 7000 |
| 18 | 24×30×6 | 6600 | 14 | 22×36 6×6 | 7200 | 18 | 22×38×6 | 7400 |

Note.—All above pans made of $\frac{3}{16}$-in. material. Ammonia condenser pan sizes given for coils on 18-in. centers.

Sizes above heavy line are built up at factory. Sizes below are shipped knocked down.

## TABLE 167

### Enclosed Type Machinery and High Sides

| | Weight, Pounds |
|---|---|
| 1 10-in.×10-in., two-cylinder, belt-driven machine | 11,400 |
| 3 Double pipe ammonia condensers, 10 P.H.×20 ft. long, 1¼ in. and 2-in. F. W. steel pipe | 5,400 |
| Gas, liquid and water headers for condensers | 324 |
| 1 Ammonia oil separator, 16 in.×4 ft. | 370 |
| 1 Ammonia receiver, 20 in.×10 ft. | 850 |
| Ammonia connections: | |
| 60 ft., 2½ in. E. H. steel pipe | |
| 5 2½ in. ammonia ells, C. F. B. & G. | |
| 2 Pr. 2½ in. ammonia flanges B. & G. | |
| 20 ft., 3 in. F. W. steel pipe | 1,250 |
| 3 3-in. ammonia ells, C. F. B. & G. | |
| 20 ft., 1 in. E. H. pipe | |
| 2 Pr. 1 in. ammonia flanges B. & G., gage connections | |
| Total | 19,594 |
| 1 10-in.×10-in., two-cylinder machine arranged for horizontal uniflow engine drive | 9,000 |
| 1 10-in.×10-in., two-cylinder machine arranged for engine type synchronous motor drive. Includes wheel, shaft extension, coupling and outboard bearing. Shaft swell permissible. No exciter pulley included. | 12,360 |
| Add to above for pulley when belted exciter is used | 500 |
| Deduct wheel at rate of 12 cents per pound list. | |
| 1 12-in.×12-in., two-cylinder, belt-driven machine | 21,300 |
| 4 Double pipe ammonia condensers, 12 P.H.×20 ft. long, 1¼-in. and 2-in. F. W. steel pipe | 8,400 |
| Gas, liquid and water headers for condensers | 448 |
| 1 Ammonia oil separator, 20 in.×5 ft. | 600 |
| 1 Ammonia receiver, 20 in.×16 ft. | 1,200 |
| Ammonia connections: | |
| 60 ft., 3½ in. E. H. steel pipe | |
| 5 3½ in. ammonia ells, C. F. B. & G. | |
| 2 Pr. 3½ in. ammonia flanges B. & G. | |
| 40 ft., 4 in. F. W. steel pipe | 2,200 |
| 3 4 in. ammonia ells, C. F. B. & G. | |
| 40 ft. 1¼ in. E. H. pipe | |
| 2 Pr. 1¼ in. ammonia flanges, B. & G., gage connections | |
| Total | 34,148 |
| 1 12-in.×12-in., two-cylinder machine arranged for horizontal uniflow engine drive | 15,300 |
| 1 12-in.×12-in., two-cylinder machine arranged for engine type synchronous motor drive. Included wheel, shaft extension, coupling and outboard bearing. Shaft swell permissible. No exciter pulley included. | 23,050 |
| Add to above for pulley when belted exciter is used | 600 |

TABLE 168

HORIZONTAL UNIFLOW ENGINES—SPECIFICATIONS

| Steam Cylinder, Inches | | Flywheel | | | Main Bearing, Inches | | Crank Pin, Inches | | Cross Head Pin, Inches | | Connecting Rod, Center to Center, in Inches | Piston Rod Diameter, Inches | Pipe Connections, Inches | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Diameter | Stroke | Diameter | | Weight, pounds | Diameter | Length | Diameter | Length | Diameter | Length | | | Steam | Exhaust |
| | | Ft. | In. | | | | | | | | | | | |
| 12 | 12 | 6 | 0 | 2,500 | 6 | 12 | $4\frac{1}{2}$ | 4 | 3 | 5 | 40 | $2\frac{5}{8}$ | $3\frac{1}{2}$ | $4\frac{1}{2}$ |
| 12 | 14 | 6 | 0 | 2,500 | 6 | 12 | $4\frac{1}{2}$ | 4 | 3 | 5 | 40 | $2\frac{5}{8}$ | $3\frac{1}{2}$ | $4\frac{1}{2}$ |
| 14 | 14 | 7 | 0 | 3,800 | 7 | 14 | 5 | 5 | $3\frac{1}{2}$ | $5\frac{1}{2}$ | 44 | 3 | 4 | 5 |
| 14 | 16 | 7 | 0 | 3,800 | 7 | 14 | 5 | 5 | $3\frac{1}{2}$ | $5\frac{1}{2}$ | 44 | 3 | 4 | 5 |
| 16 | 16 | 8 | 0 | 5,000 | 8 | 16 | 6 | 6 | 4 | $6\frac{1}{2}$ | 48 | $3\frac{1}{2}$ | 5 | 6 |
| 16 | 18 | 8 | 0 | 5,000 | 8 | 16 | 6 | 6 | 4 | $6\frac{1}{2}$ | 48 | $3\frac{1}{2}$ | 5 | 6 |
| 18 | 18 | 8 | 6 | 6,500 | 9 | 18 | 7 | 6 | $4\frac{1}{2}$ | $7\frac{1}{2}$ | 54 | $3\frac{7}{8}$ | 6 | 7 |
| 18 | 20 | 8 | 6 | 6,500 | 9 | 18 | 7 | 6 | $4\frac{1}{2}$ | $7\frac{1}{2}$ | 54 | $3\frac{7}{8}$ | 6 | 7 |
| 20 | 20 | 9 | 6 | 8,000 | 10 | 20 | 8 | 7 | 5 | 9 | 60 | $4\frac{1}{4}$ | 7 | 8 |
| 20 | 22 | 9 | 6 | 8,000 | 10 | 20 | 8 | 7 | 5 | 9 | 60 | $4\frac{1}{4}$ | 7 | 8 |
| 22 | 22 | 9 | 6 | 10,000 | 11 | 22 | 9 | 8 | 6 | $9\frac{1}{2}$ | 66 | $4\frac{3}{4}$ | 7 | 9 |
| 24 | 22 | 9 | 6 | 12,000 | 11 | 22 | 9 | 8 | 6 | $9\frac{1}{2}$ | $60\frac{1}{2}$ | $5\frac{1}{8}$ | | |
| 22 | 24 | 9 | 6 | 10,000 | 11 | 22 | 9 | 8 | 6 | $9\frac{1}{2}$ | 66 | $4\frac{3}{4}$ | 7 | 9 |
| 24 | 24 | 9 | 6 | 13,500 | 12 | 24 | | | | | | | | |
| 24 | 26 | 9 | 6 | 13,500 | 12 | 24 | | | | | | | | |
| 26 | 26 | 10 | 0 | 15,000 | 13 | 26 | | | | | | | | |
| 26 | 28 | 10 | 0 | 15,000 | 13 | 26 | | | | | | | | |
| 27 | 28 | 10 | 0 | 16,500 | $13\frac{1}{2}$ | 27 | ..... | ..... | ..... | ..... | $82\frac{1}{2}$ | | | |
| 27 | 30 | 10 | 0 | 16,500 | $13\frac{1}{2}$ | 27 | 11 | 10 | $7\frac{1}{2}$ | 11 | $82\frac{1}{2}$ | $5\frac{1}{8}$ | 10 | 12 |

Engines 24 in. $\times$ 24 in. and larger are equipped with tail rod. When engine room space will not permit use of tail rod, consult Home Office for engine recommendations. When engines are to be used with H. D. A. long stroke compressors the disengaging type valve gear with fly-ball governor will be used.

TABLE 169

FREEZING TANKS, BULKHEADS AND PARTITIONS COIL TANKS, 300 LB. STANDARD CANS

| Tons of Ice, Basis of 16 Cans per ton | Number of Cans | Standard Tank Sizes Using Horizontal Agitators. For Vertical Agitators Add 7 In. to Tank Length. Tanks 4 Ft. 0 In. Deep. Material ¼-In. Tank Steel. | | | | Weight per Square Foot | Weight, Pounds | Bulkheads and Partitions | |
|---|---|---|---|---|---|---|---|---|---|
| | | Length | | Width | | | | Number of agitators | Weight, pounds |
| | | Ft. | In. | Ft. | In. | | | | |
| 1 | 16 | 11 | 0 | 5 | 10 | 12 | 2,390 | 1 | 200 |
| | 15 | 13 | 0½ | 4 | 7½ | 12 | 2,440 | 1 | 220 |
| 2 | 30 | 15 | 1 | 7 | 0½ | 12 | 3,410 | 1 | 250 |
| | 30 | 13 | 0½ | 8 | 3 | 12 | 3,360 | 1 | 240 |
| 3 | 42 | 15 | 1 | 9 | 5½ | 12 | 4,080 | 1 | 270 |
| | 42 | 17 | 1½ | 8 | 3 | 12 | 4,150 | 1 | 280 |
| 4 | 56 | 17 | 1½ | 10 | 8 | 11½ | 4,660 | 1 | 290 |
| | 54 | 21 | 2½ | 8 | 3 | 11½ | 4,730 | 1 | 300 |
| 5 | 72 | 19 | 2 | 11 | 10½ | 11½ | 5,460 | 1 | 340 |
| | 70 | 23 | 3 | 9 | 5½ | 11½ | 5,560 | 1 | 350 |
| 5½ | 80 | 19 | 2 | 13 | 1 | 11½ | 5,860 | 1 | 360 |
| | 80 | 23 | 3 | 10 | 8 | 11½ | 5,980 | 1 | 370 |
| 6 | 99 | 21 | 2½ | 14 | 3½ | 11½ | 6,800 | 1 | 380 |
| | 99 | 25 | 3½ | 11 | 10½ | 11½ | 6,830 | 1 | 390 |
| 7½ | 120 | 23 | 3 | 15 | 6 | 11¼ | 7,550 | 1 | 400 |
| | 120 | 27 | 4 | 13 | 1 | 11¼ | 7,690 | 1 | 420 |
| 9 | 144 | 27 | 4 | 15 | 6 | 11¼ | 8,630 | 1 | 440 |
| | 144 | 35 | 6 | 11 | 10½ | 11¼ | 9,000 | 1 | 480 |
| 11 | 182 | 29 | 4½ | 17 | 11 | 11¼ | 10,240 | 1 | 500 |
| | 180 | 33 | 5½ | 15 | 6 | 11¼ | 10,260 | 1 | 520 |
| 15 | 240 | 33 | 5½ | 20 | 4 | 11¼ | 12,630 | 1 | 550 |
| | 238 | 37 | 6½ | 17 | 11 | 11¼ | 12,850 | 1 | 600 |
| 19 | 306 | 37 | 6½ | 22 | 9 | 11¼ | 15,370 | 1 | 650 |
| | 304 | 42 | 3½ | 20 | 4 | 11¼ | 15,420 | 1 | 775 |
| 22 | 360 | 39 | 7 | 25 | 2 | 11 | 17,000 | 2 | 1000 |
| | 360 | 44 | 4 | 22 | 9 | 11 | 17,090 | 1 | 900 |
| 26 | 418 | 42 | 3½ | 27 | 7 | 11 | 19,100 | 2 | 1150 |
| | 420 | 46 | 4½ | 25 | 2 | 11 | 19,240 | 2 | 1200 |
| 30 | 484 | 48 | 5 | 27 | 7 | 11 | 21,430 | 2 | 1300 |
| | 480 | 52 | 6 | 25 | 2 | 11 | 21,490 | 2 | 1350 |
| 37½ | 600 | 54 | 6½ | 30 | 0 | 11 | 25,500 | 2 | 1400 |
| | 600 | 64 | 9 | 25 | 2 | 11 | 25,890 | 2 | 1450 |
| 45 | 720 | 64 | 9 | 30 | 0 | 11 | 29,830 | 2 | 1500 |
| | 720 | 77 | 0 | 25 | 2 | 11 | 30,340 | 2 | 1550 |

For larger tonnages than those given use two or more smaller tanks of proper size.

TABLE 169*a*

FREEZING TANKS, BULKHEADS AND PARTITIONS COIL TANKS, 400 LB. STANDARD CANS

| Tons of Ice, Basis of 12 Cans per ton | Number of Cans | Standard Tank Sizes Using Horizontal Agitators. For Vertical Agitators Add 7 In. to Tank Length. Tanks 5 Ft. 1 In. Deep. Material ¼-in. Tank Steel. | | | | Weight per Square Foot | Weight, Pounds | Bulkheads and Partitions | |
|---|---|---|---|---|---|---|---|---|---|
| | | Length | | Width | | | | Number of agitators | Weight, pounds |
| | | Ft. | In. | Ft. | In. | | | | |
| 5 | 56 | 17 | 1½ | 10 | 8 | 11½ | 5,360 | 1 | 370 |
| 4 | 54 | 21 | 2½ | 8 | 3 | 11½ | 5,480 | 1 | 390 |
| 5 | 63 | 17 | 1½ | 11 | 10½ | 11½ | 5,760 | 1 | 390 |
| 5 | 63 | 21 | 2½ | 9 | 5½ | 11½ | 5,790 | 1 | 420 |
| 7 | 80 | 19 | 2 | 13 | 1 | 11¼ | 6,500 | 1 | 470 |
| 7 | 80 | 23 | 3 | 10 | 8 | 11¼ | 6,670 | 1 | 470 |
| 8 | 90 | 21 | 2½ | 13 | 1 | 11¼ | 7,060 | 1 | 500 |
| 7 | 88 | 25 | 3½ | 10 | 8 | 11¼ | 7,160 | 1 | 520 |
| 9 | 108 | 21 | 2½ | 15 | 6 | 11¼ | 7,920 | 1 | 550 |
| 9 | 108 | 27 | 4 | 11 | 10½ | 11¼ | 8,170 | 1 | 580 |
| 11 | 132 | 27 | 4 | 14 | 3½ | 11¼ | 9,180 | 1 | 600 |
| 11 | 135 | 33 | 5½ | 11 | 10½ | 11¼ | 9,710 | 1 | 700 |
| 15 | 182 | 29 | 4½ | 17 | 11 | 11¼ | 11,480 | 2 | 950 |
| 15 | 180 | 33 | 5½ | 15 | 6 | 11¼ | 11,570 | 1 | 850 |
| 20 | 224 | 31 | 5 | 20 | 4 | 11¼ | 13,220 | 2 | 1000 |
| 20 | 224 | 35 | 6 | 17 | 11 | 11 | 13,590 | 2 | 1100 |
| 20 | 270 | 33 | 5½ | 22 | 9 | 11 | 14,810 | 2 | 1200 |
| 20 | 272 | 37 | 6½ | 20 | 4 | 11 | 15,200 | 2 | 1350 |
| 25 | 320 | 35 | 6 | 25 | 2 | 11 | 17,000 | 2 | 1425 |
| 25 | 320 | 44 | 4 | 20 | 4 | 11 | 17,290 | 2 | 1550 |
| 30 | 360 | 39 | 7 | 25 | 2 | 11 | 18,590 | 2 | 1650 |
| 30 | 360 | 44 | 4 | 22 | 9 | 11 | 18,770 | 2 | 1725 |
| 40 | 460 | 50 | 5½ | 25 | 2 | 11 | 22,530 | 2 | 1800 |
| 40 | 450 | 54 | 6½ | 22 | 9 | 11 | 22,420 | 2 | 1875 |
| 45 | 540 | 58 | 7½ | 25 | 2 | 11 | 25,730 | 2 | 1950 |
| 45 | 540 | 64 | 9 | 22 | 9 | 11 | 26,120 | 2 | 2000 |

For larger tonnages than those given use two or more smaller tanks of proper size.

TABLE 170

DISTILLED WATER ICE MAKING APPARATUS

| D. P. Distilled Water Cooler | | | | | | Pipe Connections, Weight, Pounds |
|---|---|---|---|---|---|---|
| Tons of ice | Number of coils | Pipes high | Length | | Weight, pounds | |
| | | | Ft. | In. | | |
| 1 | 1 | 2 | 7 | 0 | 155 | 130 |
| 2 | 1 | 2 | 10 | 0 | 225 | 140 |
| 3 | 1 | 3 | 10 | 0 | 315 | 150 |
| 4 | 1 | 4 | 10 | 0 | 385 | 150 |
| 5 | 1 | 6 | 10 | 0 | 585 | 300 |
| 6 | 1 | 6 | 10 | 0 | 585 | 300 |
| 8 | 1 | 7 | 10 | 0 | 665 | 300 |
| 10 | 1 | 8 | 10 | 0 | 750 | 600 |
| 12 | 1 | 8 | 10 | 0 | 750 | 600 |
| 15 | 1 | 6 | 20 | 0 | 1,000 | 650 |
| 20 | 1 | 8 | 20 | 0 | 1,300 | 670 |
| 25 | 2 | 6 | 20 | 0 | 2,000 | 680 |
| 30 | 2 | 6 | 20 | 0 | 2,000 | 700 |
| 35 | 2 | 8 | 20 | 0 | 2,600 | 800 |
| 40 | 2 | 8 | 20 | 0 | 2,600 | 900 |
| 50 | 3 | 8 | 20 | 0 | 3,900 | 950 |
| 60 | 3 | 8 | 20 | 0 | 3,900 | 1000 |
| 75 | 4 | 8 | 20 | 0 | 5,200 | 1200 |
| 100 | 5 | 8 | 20 | 0 | 6,500 | 1500 |
| 125 | 7 | 8 | 20 | 0 | 9,100 | 2000 |
| 150 | 8 | 8 | 20 | 0 | 10,400 | 2400 |
| 200 | 10 | 8 | 20 | 0 | 13,000 | 3000 |

Distilled Water Filters

| Tons of Ice | Number of Filters | Size of Filters, Inches | Weight, Pounds |
|---|---|---|---|
| 1 | 1 | 24×60 | 6000 |
| 2 to 30 | 2 | 24×60 | 1200 |
| 35 to 40 | 3 | 24×60 | 1700 |
| 50 to 60 | 4 | 24×60 | 2300 |
| 75 | 5 | 24×60 | 2800 |
| 100 | 6 | 24×60 | 3350 |
| 125 | 7 | 24×60 | 3900 |
| 150 | 8 | 24×60 | 4475 |
| 200 | 10 | 24×60 | 5575 |

Distilled water coolers are made of $1\frac{1}{4}$-in. and 2-in. F. W. Galvanized pipe. Weights include cooler only. No valves nor headers.

Pipe connections include all necessary distilled water piping and fittings.

TABLE 170—*Continued*

| Tons of ice | Flask Steam Condensers | | | | Tons of ice | Flask Steam Condensers | | | |
|---|---|---|---|---|---|---|---|---|---|
| | Number of coils | Length | | Weight, pounds | | Number of coils | Length | | Weight, pounds |
| | | Ft. | In. | | | | Ft. | In. | |
| 1 | 1 | 9 | 9 | 670 | 30 | 3 | 11 | 9 | 2,390 |
| 2 | 1 | 9 | 9 | 670 | 35 | 3 | 15 | 6 | 3,170 |
| 3 | 1 | 9 | 9 | 670 | 40 | 3 | 19 | 6 | 3,915 |
| 4 | 1 | 9 | 9 | 670 | 50 | 4 | 19 | 6 | 5,120 |
| 5 | 1 | 9 | 9 | 670 | 60 | 5 | 19 | 6 | 6,530 |
| 6 | 1 | 9 | 9 | 670 | 75 | 6 | 19 | 6 | 7,840 |
| 8 | 1 | 9 | 9 | 670 | 100 | 7 | 19 | 6 | 9,150 |
| 10 | 1 | 11 | 9 | 750 | 125 | 9 | 19 | 6 | 11,750 |
| 12 | 1 | 15 | 6 | 1010 | 150 | 11 | 19 | 6 | 14,370 |
| 15 | 2 | 9 | 9 | 1420 | 200 | 14 | 19 | 6 | 18,200 |
| 20 | 2 | 11 | 9 | 1580 | | | | | |
| 25 | 2 | 15 | 6 | 2115 | | | | | |

| Tons of ice | Reboilers | | | | | Tons of ice | Reboilers | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | Size of tanks | | | | Weight, pounds | | Size of tanks | | | | Weight, pounds |
| | Ft. | In. | In. | Ft. | | | Ft. | In. | In. | Ft. | |
| 1 thru 6 | 7 | 0 | ×16 | ×18 | 440 | 35 and 40 | 17 | $\frac{5}{8}$ | ×30 | ×18 | 1350 |
| 8 thru 12 | 7 | 0 | ×30 | ×18 | 630 | 50 and 60 | 20 | 6 | ×30 | ×18 | 1550 |
| 15 and 20 | 10 | $3\frac{5}{8}$ | ×30 | ×18 | 900 | 75 and 100 | 23 | $9\frac{5}{8}$ | ×30 | ×18 | 1820 |
| 25 and 30 | 13 | 9 | ×30 | ×18 | 1100 | 125 thru 200 | 27 | 0 | ×30 | ×18 | 2170 |

Distilled Water Refrigerators

| Tons of Ice | Connections, Inches | Weight, Pounds | Tons of Ice | Connections, Inches | Weight, Pounds |
|---|---|---|---|---|---|
| 1 to 12 | $\frac{3}{4}$ | 80 | 50 to 60 | 2 | 91 |
| 15 to 20 | 1 | 83 | 75 to 100 | $2\frac{1}{2}$ | 100 |
| 25 to 30 | $1\frac{1}{4}$ | 85 | 125 to 200 | 3 | 108 |
| 35 to 40 | $1\frac{1}{2}$ | 87 | | | |

| Tons of Ice | Float Tanks | | Tons of Ice | Float Tanks | |
|---|---|---|---|---|---|
| | Size of tank, inches | Weight, pounds | | Size of tank, inches | Weight, pounds |
| 1 to 6 | 15×24 | 115 | 8 to 200 | 20×30 | 205 |

TABLE 170—*Continued*

Distilled Water Pumps

| Size of Cylinder, Inches | | Stroke, Inches | Pipe Connections, Inches | | | | Weight, Pounds |
|---|---|---|---|---|---|---|---|
| Steam | Water | | Steam | Exhaust | Suction | Discharge | |
| 3 | 2 | $3\frac{1}{2}$ | $\frac{1}{2}$ | $\frac{3}{4}$ | $1\frac{1}{4}$ | 1 | 125 |
| $3\frac{1}{2}$ | $2\frac{1}{4}$ | 4 | $\frac{1}{2}$ | $\frac{3}{4}$ | $1\frac{1}{2}$ | $1\frac{1}{4}$ | 200 |

Steam condensers are made of 10-lb. galvanized steel and are furnished complete with stands, sprinkler, troughs and steam header.

Reboilers are made of $\frac{1}{8}$-in. steel with 16-lb. steel cover and vent pipes. Steam coil and overflow and water connections included. All painted.

Float tanks and pumps used as substitute for regulators when gravity feed to water storage tank not possible. Float tanks are made of $\frac{1}{8}$-in. steel and have float control.

TABLE 171

SPECIFICATIONS ENCLOSED TYPE V. S. A. AMMONIA COMPRESSORS

| Number of Cylinders | Cylinders, Inches | | Fly Wheel | | | Crank Shaft, Inches | | Crank Pins, Inches | |
|---|---|---|---|---|---|---|---|---|---|
| | Bore | Stroke | Diameter, inches | Face, inches | Weight, pounds | Diameter | Length of bearings | Diameter | Length |
| 1 | 3 | 3 | 26 | 3½ | 180 | 1¾ | 5 | 1½ | 1¾ |
| 2 | 3 | 3 | 26 | 3½ | 180 | 1¾ | 5 | 1½ | 1¾ |
| 1 | 4 | 4 | 30 | 4½ | 300 | 2 | 5 | 2 | 2⅜ |
| 2 | 4 | 4 | 30 | 4½ | 300 | 2 | 5 | 2 | 2⅜ |
| 2 | 5 | 5 | 36 | 6½ | 600 | 2½ | 6 | 2½ | 2¾ |
| 2 | 6 | 6 | 42 | 7½ | 600 | 3 | 7 | 3 | 3¼ |
| 2 | 7 | 7 | 48 | 8½ | 1000 | 3½ | 8 | 3½ | 3¾ |
| 2 | 8 | 8 | 50 | 8½ | 1200 | 4 | 9 | 4 | 4¾ |
| 2 | 9 | 9 | 60 | 10½ | 1500 | 4½ | 10¼ | 4½ | 5¼ |
| 2 | 10 | 10 | 72 | 12½ | 1800 | 5 | 11¼ | 5 | 5¾ |
| 2 | 12 | 12 | 86 | 14½ | 3250 | 7 | 14 | 7 | 6 |

| Number of Cylinders | Piston Pins, Inches | | Connection Rods, Inches | | Pipe Connections, Inches | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Diameter | Length | Length, center to center | Diameter of bolts | Ammonia suction | Ammonia discharge | Water jacket inlet | Water jacket outlet | Steam | Exhaust |
| 1 | ⅞ | 1½ | 7½ | ½ | ¾ | ¾ | ⅜ | ⅜ | | |
| 2 | ⅞ | 1½ | 7½ | ½ | ¾ | ¾ | ½ | ½ | | |
| 1 | 1⅛ | 2 | 10 | ½ | 1 | 1 | ¾ | ¾ | 1 | 1¼ |
| 2 | 1⅛ | 2 | 10 | ½ | 1 | 1 | 1 | 1 | 1 | 1½ |
| 2 | 1¼ | 2¾ | 12½ | ⅝ | 1¼ | 1¼ | 1 | 1 | 1½ | 2 |
| 2 | 1½ | 3¼ | 15 | ¾ | 1½ | 1½ | 1¼ | 1¼ | 1½ | 2 |
| 2 | 1¾ | 3¾ | 17½ | ⅞ | 2 | 1½ | 1¼ | 1¼ | 2 | 2½ |
| 2 | 2 | 4⅜ | 20 | 1 | 2½ | 2 | 1½ | 1½ | 2½ | 3 |
| 2 | 2¼ | 4⅞ | 22½ | 1⅛ | 2½ | 2 | 1½ | 1½ | 3 | 5 |
| 2 | 2½ | 5⅜ | 25½ | 1¼ | 3 | 2½ | 2 | 2 | | |
| 2 | 3½ | 6½ | 30 | ..... | 4 | 3½ | 2 | 2 | | |

## TABLE 172

### Enclosed Type Machines and High Sides

Items

1 4-in.×4-in., two cylinder B. D. machine.
1 D. P. ammonia condenser, 4 P.H.×20 ft. long, 1¼-in. and 2-in. F. W. steel.
1 Ammonia oil separator, 8 in.×20 in. long.
1 Ammonia receiver, 6 in. diameter×8 ft. long.

Ammonia connections:

15 ft. of 1 in. E. H. steel pipe.
7 1-in. ammonia ells, screwed.
2 Pr. 1-in. ammonia flanges B. & G. Gage connections.
Machine arranged for direct engine drive.

---

1 5-in.×5-in., two cylinder B. D. machine.
1 D. P. ammonia condenser, 6 P.H.×20 ft. long, 1¼-in. and 2-in. F. W. steel.
1 Ammonia oil separator, 8 in.×20 in. long.
1 Ammonia receiver, 8 in.×8 ft. long.

Ammonia connections:

20 ft., 1¼ in. E. H. steel pipe.
7 1¼-in. ammonia ells screwed.
2 Pr. 1¼-in. ammonia flanges, B. & G. Gage connections.
Machine arranged for direct engine drive.

---

1 6-in.×6-in., two cylinder B. D. machine.
1 D. P. ammonia condenser, 8 P.H.×20 ft. long, 1¼-in. and 2-in. F. W. steel.
1 Ammonia oil separator, 12 in.×36 in.
1 Ammonia receiver, 8 in.×8 ft. long.

Ammonia connections:

20 ft., 1½-in. E. H. steel pipe.
7 1½-in. ammonia ells screwed.
2 Pr. 1½-in. ammonia flanges, B. & G. Gage connections.
Machine arranged for direct engine drive.

---

1 7-in.×7-in., two cylinder B. D. machine.
1 D. P. ammonia condenser, 10 P.H.×20 ft. long, 1¼-in. and 2-in. F. W. steel.
1 Ammonia oil separator, 12 in.×36 in.
1 Ammonia receiver, 8-in.×18 ft.

Ammonia connections:

25 ft., 1½-in. E. H. steel pipe.
5 1½-in. ammonia ells screwed.
2 2-in. ammonia ells screwed.
2 Pr. 1½-in. ammonia flanges, B. & G. Gage connections.
Machine arranged for direct engine drive.

---

Note.—The ammonia condensers listed for all sizes Enclosed Type Machine H. S. are based on average conditions. If the conditions are extreme, determine proper size condenser from capacity sheets. The gage connections furnished with 4 in.×4 in. thru 9 in.×9 in enclosed.

Type Machine High Side consists of the following: Gages L. and H. pressure
20 ft. ¼ in. E. H. steel pipe, 2 ¼ in. angle valves, screwed.
2 ¼ in. ammonia ells, screwed. 4 Prs. ¼ in. ammonia flanges B. & G.

TABLE 172—*Continued*

Items

1 8-in.×8-in., two cylinder B. D. machine.
2 D. P. ammonia condensers, 8 P.H.×20 ft. long, 1¼-in. and 2-in. F. W. steel. Gas and liquid headers.
1 Ammonia oil separator, 12 in.×36 in.
1 Ammonia receiver, 16 in.×7 ft.

Ammonia connections:

30 ft., 2 in. E. H. steel pipe.
5 2 in. ammonia ells C. F. B. & G.
2 2½ in. ammonia ells C. F. B. & G.
2 Pr. 2-in. ammonia flanges, B. & G.
10 ft., ½ in. E. H. pipe.
1 ½-in. ammonia screw tee.
1 Pr. ½-in. ammonia flanges B. & G.
Gage connections.
Machine arranged for direct engine drive.

---

1 9-in.×9-in., two cylinder B. D. machine.
2 D. P. ammonia condensers, 10 P.H.×20 ft. long, 1¼-in. and 2-in. F. W. steel. Gas and liquid headers.
1 Ammonia oil separator, 12 in.×36 in.
1 Ammonia receiver, 20 in.×7 ft.

Ammonia connections:

30 ft., 2 in. E. H. steel pipe.
5 2 in. ammonia ells, C. F. B. & G.
2 2½-in. ammonia ells, C. F. B. & G.
2 Pr. 2-in. ammonia flanges, B. & G
10 ft., ¾-in. E. H. pipe.
1 ¾-in. ammonia screw tee.
1 Pr. ¾-in. flanges, B. & G.
Gage connections.

B. D. machine with extended base plate and outboard bearing.
Machine with extended base plate arranged for direct vertical engine drive.
Machine without entended base plate arranged for direct vertical engine drive.
Machine without extended base plate arranged for horizontal uniflow engine drive.
Machine without extended base plate or flywheel but with shaft length suitable for engine type synchronous motor drive. No swell in shaft permissible. Outboard bearing included. Motor rotor must have sufficient weight to furnish necessary flywheel effect. Motor rotor must be supplied with split hub or split rotor.

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No exciter pulley included in above.
NOTE.—The gage connections furnished with 10 in.×10 in. and 12 in.×12 in. Enclosed Type Machine H. S. consist of the following:
40 ft. ¼-in. E. H. steel pipe, 2 ¼-in. ammonia valves screwed.
2 ¼-in. ammonia ells, screwed. 4 Prs. ¼-in. ammonia flanges, B. & G.

TABLE 173

Sand Filters, Weights

| Tons of Ice | Number of Filters | Size of Filters, Inches | Style | Weight, Pounds |
|---|---|---|---|---|
| 1 | 1 | 12 × 50 | C | 700 |
| 2 | 1 | 12 × 50 | C | 700 |
| 3 | 1 | 12 × 50 | C | 700 |
| 4 | 1 | 12 × 50 | C | 700 |
| 5 | 1 | 12 × 50 | C | 700 |
| 6 | 1 | 12 × 50 | C | 700 |
| 8 | 1 | 16 × 51 | C | 1,000 |
| 10 | 1 | 16 × 51 | C | 1,000 |
| 12 | 1 | 18 × 52 | C | 1,350 |
| 15 | 1 | 18 × 52 | C | 1,350 |
| 20 | 1 | 24 × 53 | C | 2,200 |
| 25 | 1 | 24 × 53 | C | 2,200 |
| 30 | 1 | 24 × 53 | C | 2,200 |
| 35 | 1 | 24 × 53 | C | 2,200 |
| 40 | 2 | 24 × 53 | C | 4,400 |
| 50 | 2 | 24 × 53 | C | 4,400 |
| 60 | 2 | 30 × 61 | I | 5,700 |
| 80 | 2 | 30 × 61 | I | 5,700 |
| 100 | 2 | 36 × 62 | I | 8,100 |
| 120 | 3 | 30 × 61 | I | 8,550 |
| 150 | 3 | 36 × 62 | I | 12,150 |
| 200 | 4 | 36 × 62 | I | 16,200 |

## TABLE 174

### Ice Freezing Cans, Specifications, Weights

| Number of Cans per Ton per 24 Hr., 19 Lb. Suction, 250 Ft. 1¼-in. Coils per Ton | | Weight of Ice Cake, Pounds | Size of Cans, Inches | | | | Space Required in Tank for Each Can, Inches | | | Weight of Can, Pounds |
|---|---|---|---|---|---|---|---|---|---|---|
| Expansion freezing system | Flooded freezing system | | Width at top | Length at top | Height over all | Gage of galvanized iron | Wide | Long | High | |
| 22 | 20¼ | 25 | 4 | 9 | 24 | 18 | 7 | 11 | 27 | 20 |
| S-17½ | 15¾ | 50 | 5 | 12 | 32 | 16 | 8 | 14 | 35 | 27 |
| 25 | 23 | 50 | 6 | 10 | 32 | 16 | 9 | 12 | 35 | 26 |
| 22 | 20¼ | 50 | 8 | 8 | 32 | 16 | 11 | 10 | 35 | 26 |
| 14½ | 13 | 60 | 5 | 14 | 32 | 16 | 8 | 16 | 35 | 29 |
| S-22 | 20¼ | 100 | 8 | 16 | 32 | 16 | 11 | 18 | 35 | 42 |
| S-23 | 21 | 200 | 11½ | 22½ | 32 | 16 | 14½ | 24½ | 35 | 60 |
| 18½ | 16½ | 200 | 14½ | 14½ | 36 | 16 | 17½ | 16½ | 39 | 55 |
| S-15½ | 14 | 300 | 11½ | 22½ | 45 | 16 | 14½ | 24½ | 48 | 70 |
| S-11½ | 10½ | 400 | 11½ | 22½ | 58 | 14 | 14½ | 24½ | 61 | 105 |

Note.—Cans marked "S" are Association Standard.

## CHAPTER XXII

## ELECTRIC MOTORS

For some time it has been considered essential that the refrigerating compressor be capable of having a widely varying rotative speed, but this is seldom demanded at the present time. When a variation of capacity is required it can be obtained by varying the number of

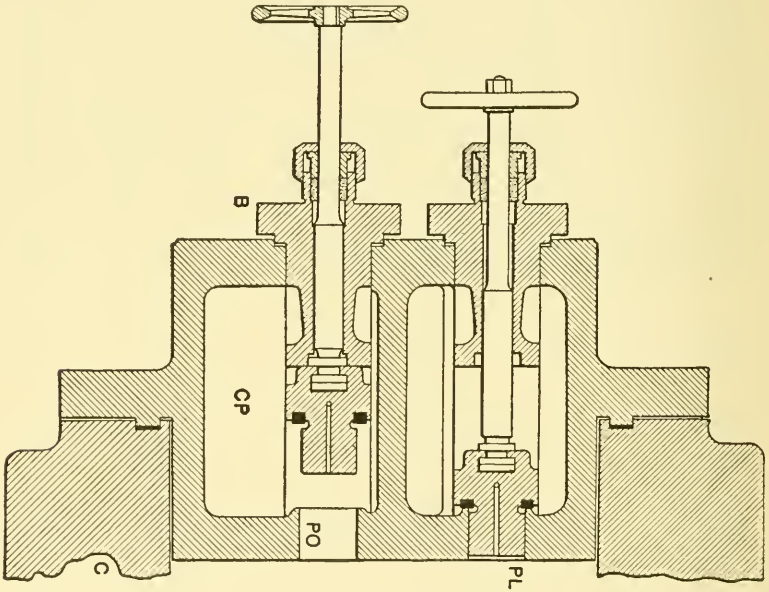


Fig. 385.—The Clearance Pocket.

machines applied to the load, varying the number of cylinders in action by disconnecting one cylinder of a duplex compressor, by increasing or decreasing the amount of clearance in the cylinder by the expedient of clearance pockets (Fig. 385), and in the smaller sizes by making the compressor automatic. In Chapter II it was shown that the amount of clearance affected very decidedly the capacity of the compressor, but

that the economy of operation, i.e., the horsepower per ton, was not appreciably affected by clearances from 4 to 8 per cent or more. Rotary speed control, therefore, is not of primary importance whereas the requirements of a good electric motor are a good starting torque and high efficiency during full speed operation. Before going into the details of the various types of motors a few electrical principles will be considered.

**Power.**—True power in an alternating electric circuit is the average value of the products of the coincident instantaneous values of the current and the voltage for a complete circuit. Power factor is the ratio of the true power, *KW*, to the apparent power, *kva*. This is expressed as

$$PF = \frac{KW}{kva}$$

The power in an electric circuit is:

$$\text{Watts} = EI \times PF \text{ for single phase}$$

$$= 2EI \times PF \text{ for two phase}$$

$$= 1.732\ EI \times PF \text{ for three phase,}$$

where $I$ = the effective line current and $E$ = the effective volts between the terminals. The effect of changes of voltage affects the starting torque as the *square* of the voltage applied to the primary.

Three-phase circuits may be connected in delta ($\Delta$) or in star (Y). If three transformers are connected in delta, the voltage across each transformer is the same as the line voltage, whereas the current in each line wire is $\sqrt{3}$ times the current in the transformer winding. If three transformers are connected in star the current will be the same as the line current, but the voltage will be $\sqrt{3}$ times the voltage across the transformer windings.

**Wire and Wiring.**—The size of conductors can be obtained by the formula, for single and two phase with unit power factor at the motor:

$$cm = \frac{21.6DI}{e},$$

where

*cm* is the size of the copper wire in circular mils;
*D* is the length of the circuit, in feet;
*I* is the current, in amperes;
*e* is the voltage drop in the length of circuit *D*.

For three phase use the constant 10.8 where *e* is the voltage drop per wire. The line voltage at the motor divided by $\sqrt{3}$ is the supply voltage minus the voltage drop *e*.

TABLE 175

DIMENSIONS, WEIGHTS AND RESISTANCES OF PURE COPPER WIRE (BARE)—BROWN AND SHARPE GAGE

| Gage Number | Diameter, Inch | Area | | Weight, Pounds per 1000 Ft. | Length, Feet per Pound | Resistance at 75 Deg. F. | | | Carrying Capacity, Amperes | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | Circular mils ($d^2$) mils = .001 inches | Square mils ($d^2$) × .7854 | | | Length, Feet per ohm | R ohms per 1000 ft. | Ohms per pound | Rubber insulation | Other insulation |
| | 1.152 | 1,000,000 | 785,400 | 3050.0 | .3275 | 95,100.0 | .01051 | .000003442 | 650 | 1000 |
| | 1.035 | 800,000.0 | 628,000.0 | 2440.0 | .4100 | 76,100.0 | .01313 | .000005380 | 550 | 840 |
| | .963 | 700,000.0 | 549,500.0 | 2135.0 | .4680 | 66,600.0 | .01501 | .000007030 | 500 | 760 |
| | .891 | 600,000.0 | 471,000.0 | 1830.0 | .5460 | 57,100.0 | .01751 | .000009579 | 450 | 680 |
| | .819 | 500,000.0 | 392,000.0 | 1525.0 | .6550 | 47,500.0 | .02101 | .00001376 | 390 | 590 |
| | .728 | 400,000.0 | 313,900.0 | 1220.0 | .8100 | 38,050.0 | .02627 | .00002155 | 330 | 500 |
| | .590 | 250,000.0 | 196,000.0 | 762.0 | 1.32 | 23,750.0 | .04203 | .00005600 | 275 | 412 |
| 0000 | .4600 | 211,600.0 | 166,190.0 | 639.3 | 1.56 | 20,383.0 | .04906 | .00007673 | 225 | 325 |
| 000 | .4096 | 167,805.0 | 131,790.0 | 507.0 | 1.97 | 16,165.0 | .06186 | .00012039 | 175 | 275 |
| 00 | .3648 | 133,079.0 | 104,520.0 | 402.0 | 2.49 | 12,820.0 | .07801 | .00009423 | 150 | 225 |
| 0 | .3248 | 105,538.0 | 82,887.0 | 318.8 | 3.14 | 10,166.0 | .09838 | .00038500 | 125 | 200 |
| 1 | .2893 | 83,694.0 | 65,733.0 | 252.8 | 3.99 | 8,062.3 | .12404 | .00048994 | 100 | 150 |
| 2 | .2576 | 66,373.0 | 52,130.0 | 200.5 | 4.99 | 6,393.7 | .15640 | .00078045 | 90 | 125 |
| 3 | .2294 | 52,634.0 | 41,339.0 | 159.0 | 6.29 | 5,070.2 | .19723 | .0012406 | 90 | 100 |
| 4 | .2043 | 41,742.0 | 32,784.0 | 126.1 | 7.93 | 4,021.0 | .24869 | .0019721 | 70 | 90 |
| 5 | .1819 | 33,102.0 | 25,998.0 | 100.0 | 10.00 | 3,188.7 | .31361 | .0031361 | 55 | 80 |
| 6 | .1620 | 26,250.0 | 20,617.0 | 79.32 | 12.61 | 2,528.7 | .39546 | .0049868 | 50 | 70 |
| 7 | .1442 | 20,816.0 | 16,349.0 | 61.90 | 15.90 | 2,005.2 | .49871 | .0079294 | 38 | 54 |

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| 8 | .1284 | 16,509.0 | 12,960.0 | 49.88 | 20.05 | 1,590.3 | .62881 | .0102608 | 35 | 50 |
| 9 | .114 | 13,094.0 | 10,284.0 | 39.56 | 25.28 | 1,261.3 | .79281 | .020042 | 28 | 38 |
| 10 | .1018 | 10,381.0 | 8,153.0 | 31.37 | 31.38 | 1,000.0 | 1.0 | .031380 | 25 | 30 |
| 11 | .0907 | 8,234.0 | 6,467.0 | 24.88 | 40.20 | 1,000.0 | 1.2607 | .0506S2 | 20 | 27 |
| 12 | .0808 | 6,529.0 | 5,128.0 | 19.73 | 50.69 | 629.02 | 1.5898 | .080585 | 20 | 25 |
| 13 | .0719 | 5,179.0 | 4,067.0 | 15.65 | 63.91 | 298.83 | 2.0047 | .12841 | 14 | 20 |
| 14 | .0640 | 4,106.0 | 3,225.0 | 12.44 | 80.38 | 395.60 | 2.5278 | .20322 | 15 | 20 |
| 15 | .0570 | 3,256.0 | 2,557.0 | 9.84 | 101.63 | 321.02 | 3.1150 | .31658 | | |
| 16 | .0508 | 2,582.0 | 2,028.0 | 7.81 | 128.14 | 248.81 | 4.0191 | .51501 | 6 | 10 |
| 17 | .0452 | 2,049.0 | 1,608.0 | 6.19 | 161.59 | 197.30 | 5.0683 | .81900 | | |
| 18 | .0403 | 1,624.0 | 1,275.0 | 4.91 | 203.76 | 156.47 | 6.3911 | 1.3023 | 3 | 5 |
| 19 | .0358 | 1,287.0 | 1,011.0 | 3.88 | 257.47 | 123.99 | 8.0654 | 2.0759 | | |
| 20 | .0319 | 1,021.0 | 802.2 | 3.09 | 324.00 | 98.40 | 10.163 | 3.2926 | | |
| 21 | .0284 | 801.1 | 636.2 | 2.45 | 408.56 | 78.06 | 12.815 | 5.2355 | | |
| 22 | .0253 | 642.7 | 504.7 | 1.94 | 515.15 | 61.91 | 16.152 | 8.3208 | | |
| 23 | .0225 | 509.4 | 500.1 | 1.51 | 649.66 | 49.08 | 20.377 | 13.238 | | |
| 24 | .0201 | 404.0 | 317.3 | 1.22 | 819.21 | 38.01 | 25.695 | 21.050 | | |
| 25 | .0179 | 320.4 | 251.6 | .97 | 1,032.9 | 30.86 | 32.400 | 33.466 | | |
| 26 | .0159 | 254.0 | 199.5 | .77 | 1,302.6 | 24.46 | 40.868 | 55.235 | | |
| 30 | .0100 | 100.5 | 78.9 | .30 | 3,293.9 | 9.681 | 103.30 | 340.25 | | |
| 36 | .0050 | 25.0 | 19.6 | .08 | 13,238.0 | 2.408 | 415.24 | 5,497.4 | | |
| 40 | .0031 | 9.9 | 7.7 | .03 | 33,157.0 | .952 | 1049.7 | 34,823.0 | | |

In lighting work no smaller than No. 14 is used, except for fixtures.

For sizes above No. 0000 the table refers to standard copper cable. Due to the conditions imposed by proper "laying" of the strands of figure for circular mileage, diameter, resistance, etc., can be given with only approximate accuracy.

One foot of pure copper wire of one circular mil cross-sectional area (diameter .001-in.) has a resistance of 10.74 ohms at 15 deg. C. or 59 deg. F.

The copper wire table (Table 175) has simple relations for approximate accuracy. A copper wire three sizes larger (B. and S. gage) than another wire has half the resistance, twice the weight and twice the area. A wire which is ten sizes larger than another has one-tenth the resistance, ten times the weight and ten times the area. The larger sized wire is specified in circular mils, and the number of feet per ohm is one-tenth of the circular mileage. To find the weight in pounds, drop four ciphers from the number of circular mils, and multiply by the weight of the No. 10 wire.

**The Direct Current Motor.**—The direct current motors are classified according to the field windings as shunt, compound and series motors. In the shunt wound motors the field is excited by connecting its winding with a field rheostat in series with the supply line. As the load increases the demand for current increases and a slight decrease in speed results in shunt motors of good design.

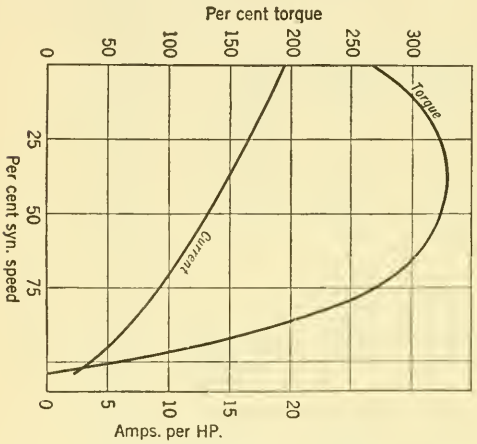

Fig. 386.—Repulsion Motors.

The *series wound* motor has its field winding in series with the armature winding, and all of the current passes through the field coils. The motor speed will vary inversely as some power of the motor load, and these motors are always direct connected by coupling, chain or gear to the load, and even then they are always under the control of the operator. They are capable of starting and accelerating heavy loads with lower current consumption than any other type of motor, and they are especially adapted for use on cranes, hoists, etc.

The cumulative *compound* wound motor is, as the name implies, a combination of both shunt and series windings for the field excitation. These motors are not constant speed, but they are used where the required torque varies considerably, being high at starting but where at the same time the speed limiting characteristics of the shunt motor are desirable.

**The Alternating Current Motors.**—In general, the alternating current motors are classified as induction, repulsion-induction and synchronous. The alternating current induction motors are classified accord-

ing to the secondary windings as squirrel cage or wound rotor. At the present times there is no satisfactory adjustable speed alternating current motor in the sense of the direct current motor, and no alternating current motor should be placed on loads requiring a variable speed.

The drop of speed in the induction motor from no load to full load is called the "slip," and this is proportional to the amount of resistance in the motor windings (Table 176). If a motor has a small slip it is considered a constant speed motor, and it compares with the direct current shunt wound motor in its speed torque characteristics. If considerable resistance is placed in the armature the slip becomes comparatively high and the speed torque characteristics are comparable with those of the compound or even with the series wound direct current motors, thereby securing a motor more adaptable for varying speed work.

Low speed motors, having a slip of from 2 to 5 per cent, of either the so-called squirrel or the wound rotor type are adapted to constant speed work, and the selection of the squirrel cage or the wound rotor motor depends on the requirements as regards the starting torque. The low slip squirrel cage motor usually has starting torques not less than the following, with full line voltage applied:

2 and 4 pole motors have 150 per cent of full load torque;
6 pole motors have 135 per cent of full load torque;
8 pole motors have 125 per cent of full load torque;
10 pole motors have 120 per cent of full load torque;
12 pole motors have 115 per cent of full load torque;
14 pole motors have 110 per cent of full load torque.

It requires from 3 to $4\frac{1}{2}$ times the full load current from the line to develop full load torque with the reduced voltage applied at starting, in the case of the squirrel cage, and only $1\frac{1}{4}$ times the full load current with the wound rotor when the proper resistance is inserted between the collector rings. The maximum running torque, or the so-called pull out torque, for all polyphase induction motors with rated impressed voltage is not less than 200 per cent of full load torque.

Squirrel cage motors of somewhat higher slip, from 8 to 12 per cent, are used where the starting torque required is large compared with the running torque, or where flywheels are employed. By varying the external resistance on a wound rotor motor a condition can be created which will practically duplicate the speed torque characteristics of the various classes of squirrel cage motors, but the current is approximately the same for all speeds.

The single phase *repulsion induction* constant speed motor is good for small capacities where a high starting torque is required as well as the lowest possible starting current. Such motors give starting torques of from 2 to 4 times the full load torque, and the maximum running torque is from $1\frac{3}{4}$ to 2 times or more the full load torque.

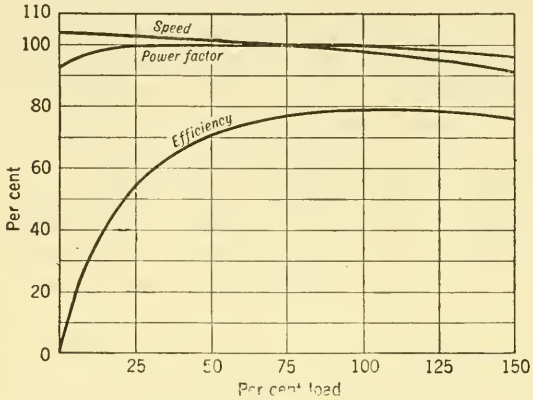


Fig. 387.—Repulsion Motors.

Figure 386 shows the operating characteristics of this type of motor, and Fig. 387 gives the efficiency and the power factor. The power factor may be made leading at light loads but at a loss of efficiency so that leading power factor is seldom called for.

**Direct Connected Electric Motors.**—In contrast to the ammonia compressor the direct connected carbonic compressor is a problem by itself due to the unbalance occasioned by a difference of about 25 per cent in the net area of the head and crank ends and the very high suction pressures of 300 to 500 lb. per sq. in. According to McLenegran[1] a double-acting, single-cylinder carbonic compressor will require

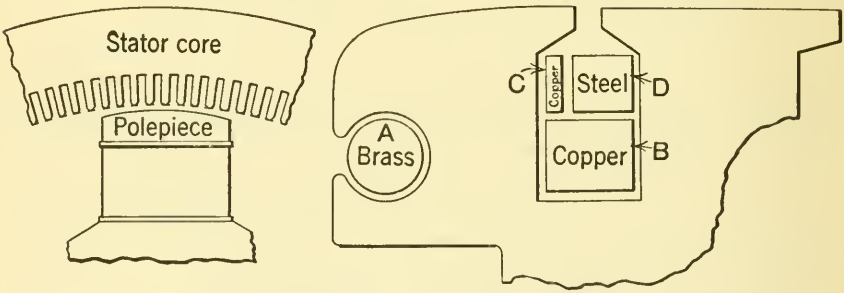


Fig. 388.—Pole Faces of Synchronous Motors.

$37\frac{1}{2}$ per cent of full load torque simply to overcome the unbalance during the starting period. After full speed is attained the flywheel will partly smooth out the unbalance.

[1] D. W. McLenegran, Refrigerating Engineering, January, 1927.

If the double-acting compressor is one with two cylinders with cranks at 90 degrees, the unbalance will be reduced to about 21 per cent, whereas the duplex double-acting compressor reduces the unbalance to zero provided the cranks are at 180 degrees.

**Flywheel Requirements.**—Considerable interest has been shown on the subject of flywheel requirements for electric driven refrigerating machines.[2] The problem appears to be one of reducing the current pulsations to a minimum, to get the required flywheel effect in the rotor of the motor if possible and to have a flywheel, or its equivalent, which will come into step easily on excitation. Stevenson has worked out a number of curves for particular designs of compressors which reduce the labor in the calculation of the required size of the flywheel.

**The Synchronous Motor.**—A number of factors have been responsible for the popularity of the synchronous motor. It lends itself to direct connection having a comparatively large air gap with respect to the induction motor (which has from 0.02 in. to 0.05 in.) and so it can take an appreciable wear of the bearings without danger of rubbing of the stator and rotor. The synchronous motor can give unity power factor or even a leading power factor without any more difficulty than an over excitation of the field. Depending on the type of compressor, the necessary flywheel effect for satisfactory operation may be incorporated in the rotor of the slow speed synchronous motor, thereby saving in space and the cost of material. It is both feasible and desirable also to throw the motor direct on the line, thus eliminating auto transformers and tandem switches, for small capacities only.

The synchronous motor, from its name, means no slip whatsoever. The torque is produced by a slight lag, some $3\frac{1}{2}$ to 6 electrical degrees, which enables it to carry the load in a manner similar to the action of torsion in a steel shaft. Such motors to be self-starting must come up to the neighborhood of synchronous speed by means of an induction motor feature in the design. This consists of brass or copper bars, or iron or steel conductors inside brass tubes, placed in slots in the pole faces, the bars being connected together by means of end rings. During the starting period the direct current excitation is removed, and the revolving flux produced by the alternating current in the armature windings reacts on the squirrel cage windings as in the induction motor. As the induction motor cannot reach synchronous speed the final

[2] Theodore Schou, The Present Status of Synchronous Motors for Direct Connection to Compressors, Amer. Society Refrigerating Engineers, May 26, 1922.

A. R. Stevenson, The Flywheel Problem in Compressors Direct Connected to Synchronous Motors, Refrigerating Engineering, October, 1924.

"pull in" is obtained when the direct current excitation is applied, which it will do if the speed is within 2 to 5 per cent of synchronism, depending on the kind of load and the amount of flywheel effect which it is necessary to accelerate. In the case of reciprocating compressors, which give a varying resisting torque, there is a periodic variation in the displacement angle of the rotor, and a corresponding fluctuation in the current drawn from the supply. To make this variation a minimum,

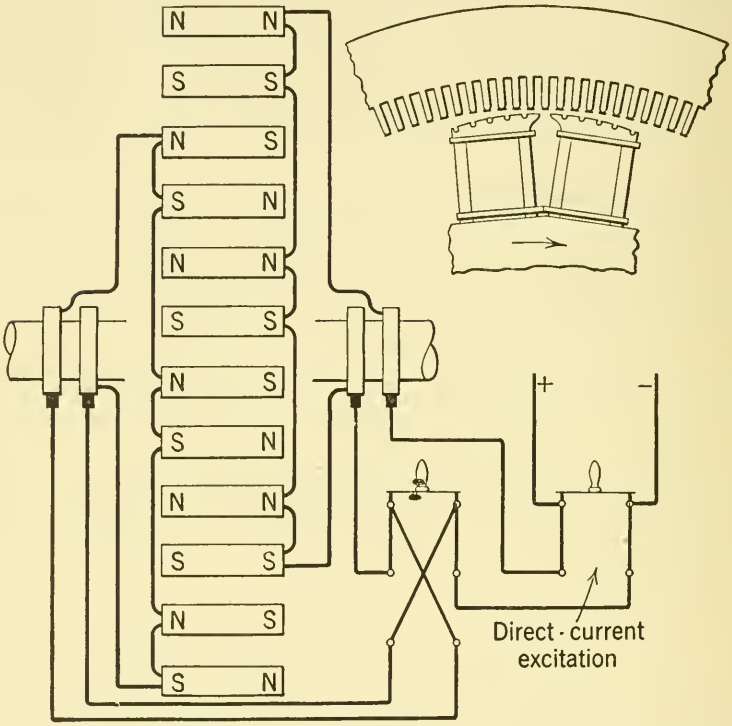


Fig. 389.—Two Speed Synchronous Motor.

the weight of the flywheel should be sufficient, but too great a weight will prevent a proper starting of the motor.

Figure 388 shows the arrangement of one type of pole-face windings. About 35 per cent of the full load torque is required to start compressors unloaded. Torques of 40 per cent will bring the average compressor up to 95 per cent synchronous speed in less than 30 seconds. The modern type of motor will deliver not less than 140 per cent of the full load torque before falling out of synchronism. The modern tendency to decrease the air gap decreases the required amount of direct current excitation.

Figure 389 shows a design of a two-speed synchronous motor. At times such a variation of speed is desirable, and the efficiency of such a motor operating at speeds of, say, 600 and 300 r.p.m. in the medium sizes, is about 95 per cent at both speeds.

TABLE 176

SPEED OF ROTARY FIELD FOR DIFFERENT NUMBERS OF POLES AND FOR VARIOUS FREQUENCIES

| Number of Poles | Speed of Revolving Magnetism, in R.p.m., When Frequency Is | | | | | |
|---|---|---|---|---|---|---|
| | 25 | 30 | $33\frac{1}{3}$ | 40 | 50 | 60 |
| 2 | 1500 | 1870 | 2000 | 2400 | 3000 | 3600 |
| 4 | 750 | 900 | 1000 | 1200 | 1500 | 1800 |
| 6 | 500 | 600 | 667 | 800 | 1000 | 1200 |
| 8 | 375 | 450 | 500 | 600 | 750 | 900 |
| 10 | 302 | 360 | 400 | 480 | 600 | 720 |
| 12 | 250 | 300 | 333 | 400 | 500 | 600 |
| 14 | 214 | 257 | 286 | 343 | 428 | 514 |
| 16 | 188 | 225 | 250 | 300 | 375 | 450 |
| 18 | 167 | 200 | 222 | 267 | 333 | 400 |
| 20 | 150 | 180 | 200 | 240 | 300 | 360 |
| 22 | 136 | 164 | 182 | 217 | 273 | 327 |
| 24 | 125 | 150 | 167 | 200 | 250 | 300 |

SLIP OF INDUCTION MOTORS

| Capacity of Motor Horse Power | Slip, at Full Load, Per Cent | | Capacity of Motor Horse Power | Slip, at Full Load, Per Cent | |
|---|---|---|---|---|---|
| | Usual limits | Average | | Usual limits | Average |
| $\frac{1}{8}$ | 20–40 | 30 | 15 | 5–11 | 8 |
| $\frac{1}{4}$ | 10–30 | 20 | 20 | 4–10 | 7 |
| $\frac{1}{2}$ | 10–20 | 15 | 30 | 3–9 | 6 |
| 1 | 8–20 | 14 | 50 | 2–8 | 5 |
| 2 | 8–18 | 13 | 75 | 1–7 | 4 |
| 3 | 8–16 | 12 | 100 | 1–6 | $3\frac{1}{2}$ |
| 5 | 7–15 | 11 | 150 | 1–5 | 3 |
| $7\frac{1}{2}$ | 6–14 | 10 | 200 | 1–4 | $2\frac{1}{2}$ |
| 10 | 6–12 | 9 | 300 | 1–3 | 2 |

## TABLE 177

### Wire Capacity of Conduits

| Size of Wire B. and S. Gage | Diameter Rubber Insul. Double Braid in 32nds. In. | Number of Wires in One Conduit Size of Unlined, Wrought-iron, Conduit for 0–600 Volts, Rubber Insulated Double-braid Wires | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
| 14 | 8 | $\frac{1}{2}$ | $\frac{1}{2}$ | $\frac{3}{4}$ | $\frac{3}{4}$ | 1 | 1 | 1 | 1 | $1\frac{1}{4}$ |
| 12 | 9 | $\frac{1}{2}$ | $\frac{3}{4}$ | $\frac{3}{4}$ | $\frac{3}{4}$ | 1 | 1 | $1\frac{1}{4}$ | $1\frac{1}{4}$ | $1\frac{1}{4}$ |
| 10 | 10 | $\frac{1}{2}$ | $\frac{3}{4}$ | 1 | 1 | 1 | $1\frac{1}{4}$ | $1\frac{1}{4}$ | $1\frac{1}{4}$ | $1\frac{1}{4}$ |
| 8 | 11 | $\frac{1}{2}$ | 1 | $1\frac{1}{4}$ | $1\frac{1}{4}$ | $1\frac{1}{4}$ | $1\frac{1}{4}$ | $1\frac{1}{4}$ | $1\frac{1}{2}$ | $1\frac{1}{2}$ |
| 6 | 14 | $\frac{1}{2}$ | 1 | $1\frac{1}{4}$ | $1\frac{1}{4}$ | $1\frac{1}{2}$ | $1\frac{1}{2}$ | $1\frac{1}{2}$ | 2 | 2 |
| 3 | 15 | $\frac{3}{4}$ | $1\frac{1}{4}$ | $1\frac{1}{4}$ | $1\frac{1}{4}$ | $1\frac{1}{2}$ | 2 | 2 | 2 | $2\frac{1}{2}$ |
| 4 | 16 | $\frac{3}{4}$ | $1\frac{1}{4}$ | $1\frac{1}{4}$ | $1\frac{1}{2}$ | $1\frac{1}{2}$ | 2 | 2 | 2 | $2\frac{1}{2}$ |
| 3 | 17 | $\frac{3}{4}$ | $1\frac{1}{4}$ | $1\frac{1}{4}$ | $1\frac{1}{2}$ | 2 | 2 | 2 | $2\frac{1}{2}$ | $2\frac{1}{2}$ |
| 2 | 19 | $\frac{3}{4}$ | $1\frac{1}{2}$ | $1\frac{1}{2}$ | 2 | 2 | $2\frac{1}{2}$ | $2\frac{1}{2}$ | $2\frac{1}{2}$ | 3 |
| 1 | 21 | 1 | $1\frac{1}{2}$ | 2 | 2 | $2\frac{1}{2}$ | $2\frac{1}{2}$ | 3 | 3 | 3 |
| 0 | 23 | 1 | 2 | 2 | 2 | $2\frac{1}{2}$ | 3 | 3 | 3 | $3\frac{1}{2}$ |
| 00 | 24 | 1 | 2 | 2 | $2\frac{1}{2}$ | $2\frac{1}{2}$ | 3 | 3 | 3 | $3\frac{1}{2}$ |
| 000 | 26 | $1\frac{1}{4}$ | 2 | $2\frac{1}{2}$ | $2\frac{1}{2}$ | 3 | 3 | $3\frac{1}{2}$ | $3\frac{1}{2}$ | $3\frac{1}{2}$ |
| 0000 | 28 | $1\frac{1}{4}$ | 2 | $2\frac{1}{2}$ | 3 | 3 | $3\frac{1}{2}$ | $3\frac{1}{2}$ | $3\frac{1}{2}$ | |
| 200,000 C. M. | 29 | $1\frac{1}{4}$ | 2 | $2\frac{1}{2}$ | $2\frac{1}{2}$ | | | | | |
| 250,000 " | 30 | $1\frac{1}{4}$ | $2\frac{1}{2}$ | $2\frac{1}{2}$ | 3 | | | | | |
| 300,000 " | 32 | $1\frac{1}{2}$ | $2\frac{1}{2}$ | 3 | 3 | | | | | |
| 350,000 " | 1–2 | $1\frac{1}{2}$ | 3 | 3 | $3\frac{1}{2}$ | | | | | |
| 400,000 " | 1–3 | $1\frac{1}{2}$ | 3 | 3 | $3\frac{1}{2}$ | | | | | |
| 450,000 " | 1–5 | $1\frac{1}{2}$ | 3 | 3 | $3\frac{1}{2}$ | | | | | |
| 500,000 " | 1–6 | $1\frac{1}{2}$ | 3 | 3 | $3\frac{1}{2}$ | | | | | |

*Two-phase vs. Three-phase.*—Due to the superior distribution of the stator windings, three-phase motors are slightly more efficient at all loads than two-phase motors of corresponding size. The power-factor is also higher especially at light loads, and the starting torque with full-load current is greater. Furthermore, for equal conditions of load and voltage, the amount of copper required in the distributing system is less. Consequently, whenever service conditions will permit, three-phase motors are preferable to two-phase.

## TABLE 178

### Wire Sizes for Alternating Current Motors

Sizes of wire, fuses and switches for use with squirrel cage continuous rated induction motors. Voltage drop not considered. Use next larger wire for motors below 600 r.p.m. Use last column for wound rotor motors.

| Horse Power | Average Ampere Rating | Wire Size Code | Switch Amperes | Starting Fuse | Running Fuse | Wire Size Based on 125 Per Cent Load |
|---|---|---|---|---|---|---|
| 220-volt Motors, 3-phase, 60 Cycles | | | | | | |
| 1 | 3.5 | 14 | 15 | 10 | 5 | 14 |
| 3 | 9.5 | 12 | 25 | 25 | 15 | 14 |
| 5.0 | 15.4 | 10 | 50 | 35 | 20 | 12 |
| 7.5 | 22.4 | 8 | 50 | 45 | 30 | 8 |
| 10.0 | 29.0 | 6 | 75 | 65 | 40 | 7 |
| 15.0 | 42.5 | 4 | 100 | 100 | 60 | 5 |
| 20.0 | 55.0 | 2 | 150 | 130 | 75 | 4 |
| 25.0 | 68.0 | 1 | 150 | 150 | 95 | 2 |
| 30.0 | 80.0 | 0 | 200 | 200 | 100 | 1 |
| 40.0 | 105.0 | 00 | 300 | 225 | 150 | 00 |
| 50.0 | 130.0 | 000 Mcm. | 300 | 250 | 175 | 000 Mcm. |
| 75.0 | 192.0 | 300 | 400 | 400 | 275 | 250 |
| 100.0 | 252.0 | 450 | 500 | 550 | 350 | 400 |
| 150.0 | 368.0 | 800 | 800 | 850 | 550 | 600 |
| 200.0 | 484.0 | 1000 | 1000 | 1150 | 725 | 900 |
| 440-volt Motors, 3-phase, 60 Cycles | | | | | | |
| 1.0 | 1.8 | 14 | 15 | 5 | 3 | 14 |
| 3.0 | 41.8 | 14 | 15 | 15 | 7 | 14 |
| 5.0 | 7.8 | 14 | 25 | 20 | 10 | 14 |
| 7.5 | 11.0 | 10 | 25 | 25 | 15 | 14 |
| 10.0 | 15.0 | 9 | 50 | 35 | 20 | 12 |
| 15.0 | 21.0 | 8 | 50 | 50 | 30 | 10 |
| 20.0 | 27.0 | 7 | 75 | 75 | 40 | 8 |
| 25.0 | 34.0 | 6 | 75 | 75 | 50 | 7 |
| 30.0 | 40.0 | 5 | 75 | 90 | 55 | 6 |
| 40.0 | 52.0 | 3 | 150 | 135 | 75 | 4 |
| 50.0 | 65.0 | 2 | 150 | 175 | 90 | 3 |
| 75.0 | 96.0 | 0 | 300 | 225 | 140 | 0 |
| 100.0 | 126.0 | 000 Mcm. | 300 | 275 | 175 | 00 |
| 150.0 | 184.0 | 250 | 400 | 400 | 275 | 0000 Mcm. |
| 200.0 | 242.0 | 400 | 500 | 550 | 350 | 400 |

## CHAPTER XXIII

## STEAM AND OIL ENGINES

Previous to 1915, generally speaking, and 1910 in particular, the steam engine was almost the only source of power in ice making plants. At the present time (1927) steam is seldom used except in special cases, such as the packing plants, some isolated ice-making plants, especially if the water available at the plant requires distillation in order to make it fit for can ice making, and in the majority of the larger hotels and apartment houses.

Whereas in the earlier days, when coal was comparatively cheap, the Corliss engine was used almost exclusively, now it is usual to use the more economical forms of prime movers. The exceptions would be in cases such as hotel refrigeration where a heat balance is attempted and the exhaust steam is used for heating purposes in the winter and an electric motor drive is used at times when steam is not required for heating.

When economy of steam is required, and a relatively small engine is to be used, the poppet valve and the uniflow steam engines are the usual types which are likely to be selected.

**The Uniflow Steam Engine.**—One of the great losses in economy in the steam engine is that due to cylinder condensation and re-evaporation. These losses are occasioned by the absorption of heat by the cylinder walls, ports and the piston head. When saturated steam is used the exposed metal causes a partial condensation of the steam up to the point of cut-off—the walls and ports being heated accordingly—and then during the expansion of the steam after the point of cut-off these same metals begin to heat and re-evaporate the steam as soon as the steam temperature falls below that of the cylinder walls. As heat is taken from the cylinder walls and head its temperature falls to a point that is relatively cold as compared with the incoming steam at the beginning of the next stroke. The net result of this heat exchange between the steam and the metals of the engine is the use of much more steam than is required for the work performed. In order to overcome these losses it is possible to use superheated steam or a design, as in the uniflow engine, which does not permit the cooling of the engine to

develop to the extent that it does in the slide valve or the usual four-valve engines. In the uniflow design the steam is exhausted through ports in the cylinder barrel which are uncovered by the piston at the end of its stroke. The piston has to be especially long—slightly less than the length of the stroke. The uniflow engine is usually designed to operate

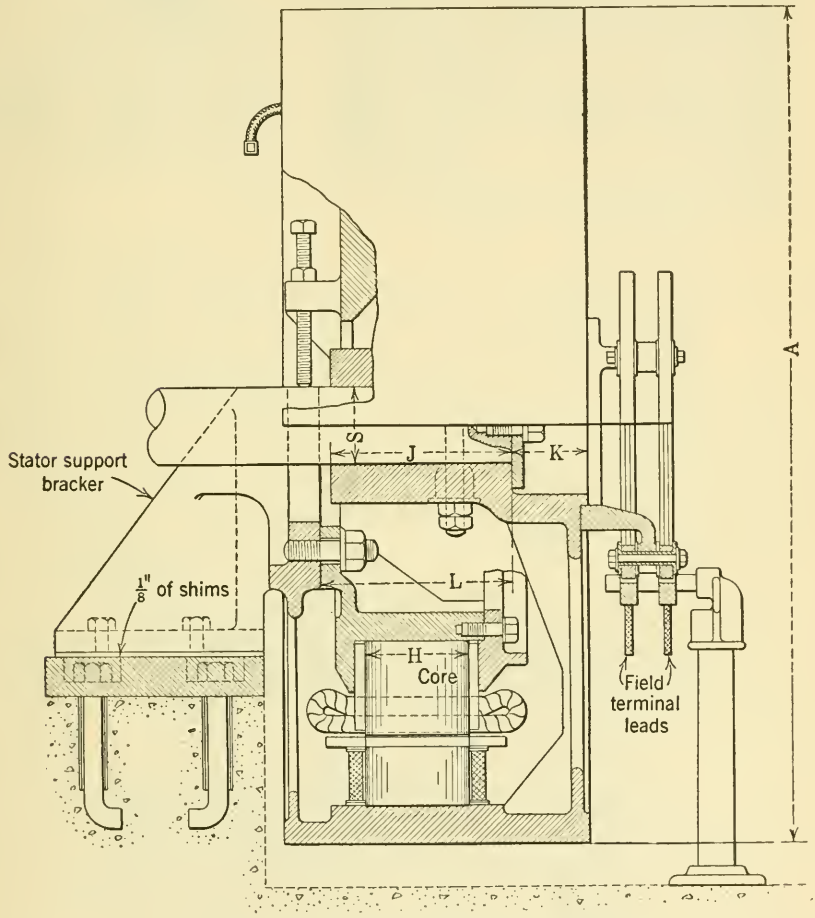


Fig. 390.—Flywheel Type Synchronous Motor.

both condensing and non-condensing, and by the use of a special device the engine may operate either manner automatically.

**The Water Rate.**—The performance curves (Figs. 392 and 395) show that with saturated steam the water rate is less than 14 lb. per i.hp.hr., but this steam consumption rate is much improved with the use of

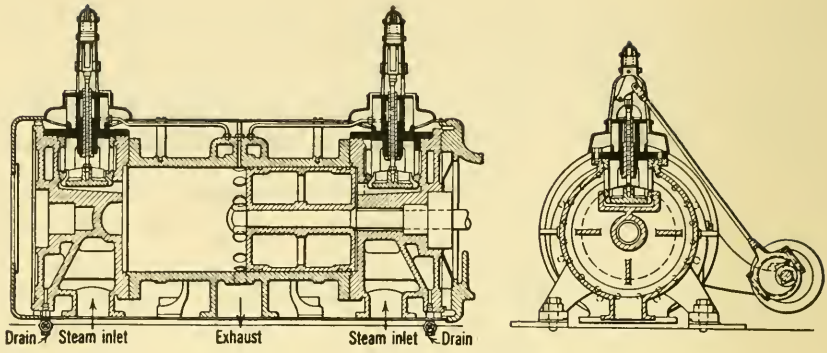


FIG. 391.—Uniflow Steam Engine.

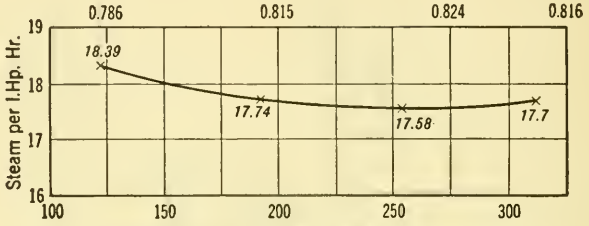


FIG. 392.—Water Rate Uniflow Engine.

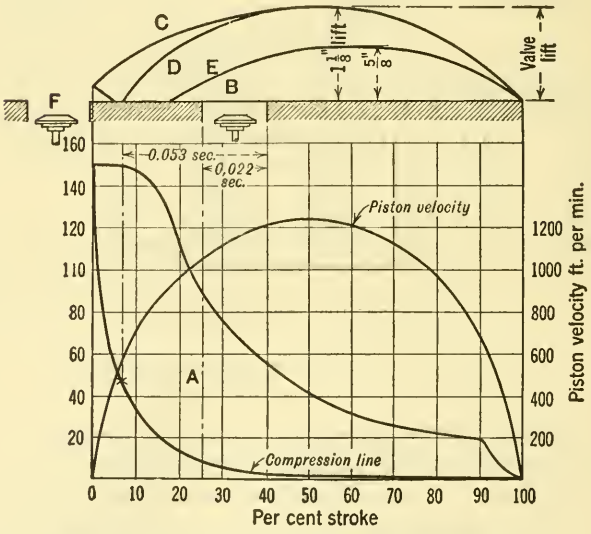


FIG. 393.—Piston Velocity of Uniflow Engine.

superheated steam. At full load and with 100 deg. F. superheat the water rate is less than 10.5 lb., and with 200 deg. superheat it is less than 9 lb. The action of the superheat is to tend to prevent the condensation of the steam, and the rate of absorption of heat by the materials of the engine is reduced very greatly on account of the lower value of the coefficient of heat transfer from metals to a gas than to a wet vapor. Sizes of uniflow steam engines are given in Tables 180 and 183.

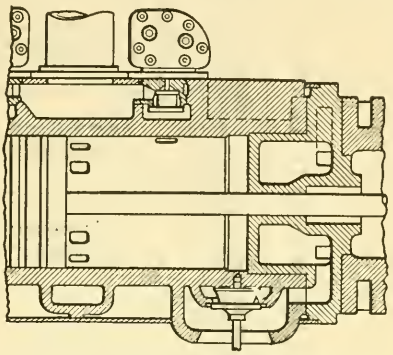

Fig. 394.—Double Seated Exhaust Uniflow Steam Engine.

**The Oil Engine.**—One of the recent outstanding developments in prime movers applied to the refrigerating industry has been that in regard to the oil engine of the Diesel and the semi-Diesel types. The economy of the Diesel engine is remarkable, being as low as 0.4 lb. of fuel oil per b.hp.hr. This type of engine has become reliable, and the maintenance cost is claimed to be less than that of the majority of small steam plants. One factor, however, is that of the operator who is not as numerous as the steam engineer, and this might cause considerable annoyance in isolated plants, except in the case of the simpler designs of semi-Diesel, hot bulb, etc., types, which are becoming quite reliable.

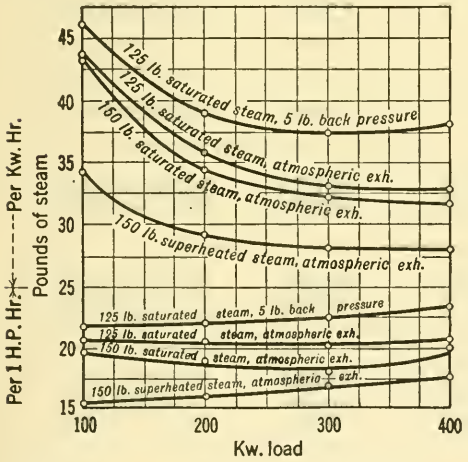


Fig. 395.—Steam Consumption Vertical Uniflow Steam Engine.

Generally speaking, it can be said that the oil engine usually employed for refrigerating plants is direct connected, although the belt drive is still found. The oil engine may be horizontal or vertical, four-cycle or two-cycle, air injection or airless injection of the fuel, full or semi-Diesel, all combined with various methods of cooling, and with details of the

## TABLE 179

TABULATION OF APPROXIMATE DIMENSIONS—SIZES, RATINGS AND STEAM CONSUMPTION AT 125 LB. STEAM

| Center Crank Flat Guide, Kw. | R.p.m. | Cylinder, Inches | Total Weight, Pounds | Shaft Weight, Pounds | Length | | Width | | Cubic Feet in Foundation | Steam Rate per 1 Hp. Hr. at 125 Lb. Saturated Steam, Atmospheric Exhaust—Proportional Load in Kw. | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | | | $\frac{1}{4}$ | $\frac{1}{2}$ | $\frac{3}{4}$ | $\frac{4}{4}$ | $\frac{5}{4}$ |
| | | | | | Ft. | In. | Ft. | In. | | | | | | |
| 150 | 277 | 12×13 | 15,500 | 1025 | 12 | 1 | 8 | 7 | 350 | 21.5 | 20.1 | 19.8 | 20.3 | 21.8 |
| 60 | 277 | 12×13 | 15,500 | 1025 | 12 | 1 | 8 | 7 | 350 | 21.0 | 19.9 | 19.9 | 21.2 | 23.6 |
| 75 | 257 | 14×15 | 22,000 | 2100 | 13 | 9 | 9 | 9 | 590 | 21.2 | 19.8 | 19.5 | 20.1 | 21.7 |
| 100 | 257 | 15×16 | 23,500 | 2100 | 14 | 2 | 10 | 7 | 710 | 20.7 | 19.5 | 19.4 | 20.4 | 22.3 |
| 125 | 257 | 16×16 | 24,000 | 2100 | 14 | 2 | 10 | 7 | 710 | 20.5 | 19.5 | 19.6 | 21.1 | 23.6 |
| 150 | 225 | 17×18 | 28,500 | 2500 | 15 | 6 | 11 | 1 | 960 | 20.2 | 19.3 | 19.7 | 21.7 | 24.4 |
| 175 | 225 | 19×20 | 37,000 | 3350 | 17 | 0 | 12 | 5 | 1130 | 20.7 | 19.3 | 19.4 | 20.2 | 22.1 |
| For 3 lb. B. P. | add..... | ..... | ..... | ..... | ..... | | ..... | | ..... | 1.4 | 1.3 | 1.2 | 1.1 | 1.1 |
| Side Crank Flat Guide, Kw. | | | | | | | | | | | | | | |
| 50 | 277 | 12×13 | 13,500 | 1200 | 12 | 0 | 9 | 8 | 430 | 21.5 | 20.1 | 19.8 | 20.3 | 21.8 |
| 60 | 277 | 12×13 | 13,500 | 1200 | 12 | 0 | 9 | 8 | 430 | 21.0 | 19.9 | 19.9 | 21.2 | 23.6 |
| 75 | 257 | 14×15 | 18,000 | 1700 | 13 | 4 | 11 | 0 | 580 | 21.2 | 19.8 | 19.5 | 20.1 | 21.7 |
| 100 | 257 | 15×16 | 21,000 | 2800 | 13 | 9 | 11 | 0 | 740 | 20.7 | 19.5 | 19.4 | 20.4 | 22.3 |
| 125 | 257 | 16×16 | 22,000 | 2800 | 13 | 9 | 11 | 0 | 740 | 20.5 | 19.5 | 19.6 | 21.1 | 23.6 |
| Side Crank Bored Guide, Kw. | | | | | | | | | | | | | | |
| 100 | 257 | 16×16 | 22,500 | 2800 | 13 | 9 | 11 | 0 | 740 | 21.3 | 19.8 | 19.4 | 19.8 | 21.2 |
| 125 | 257 | 17×16 | 22,500 | 2800 | 13 | 9 | 11 | 0 | 740 | 20.8 | 19.5 | 19.4 | 20.2 | 22.1 |
| 125 | 225 | 17×18 | 31,000 | 3700 | 16 | 5 | 12 | 6 | 1300 | 20.7 | 19.4 | 19.3 | 20.1 | 22.0 |
| 150 | 225 | 17×20 | 32,000 | 3700 | 17 | 0 | 12 | 6 | 1300 | 20.4 | 19.3 | 19.3 | 20.7 | 23.0 |
| 175 | 225 | 19×20 | 36,000 | 3700 | 17 | 0 | 12 | 6 | 1300 | 20.5 | 19.3 | 19.3 | 20.2 | 22.2 |
| 175 | 225 | 19×20 | 38,200 | 4300 | 17 | 0 | 13 | 1 | 1340 | 20.5 | 19.3 | 19.3 | 20.2 | 22.2 |
| 175 | 200 | 21×20 | 38,700 | 4300 | 17 | 0 | 13 | 1 | 1340 | 20.9 | 19.4 | 19.2 | 19.8 | 21.6 |
| 175 | 200 | 19×21 | 39,700 | 4300 | 20 | 4 | 13 | 1 | 1340 | 20.3 | 19.2 | 19.3 | 20.9 | 23.2 |
| 200 | 200 | 21×21 | 40,200 | 4300 | 20 | 4 | 13 | 1 | 1340 | 20.5 | 19.3 | 19.3 | 20.2 | 22.1 |
| 200 | 180 | 21×21 | 42,500 | 4300 | 20 | 4 | 13 | 1 | 1340 | 20.3 | 19.3 | 19.6 | 21.0 | 23.4 |

## TABLE 180

### Specifications of Anderson Diesel Engines (Type KD)—Heavy Duty

R.p.m. = 257. Piston Speed = 771 Ft. per Minute.
6-cylinder Engines Have One Flywheel

| Horse Power Rating | Kilowatt Capacity | Number of Cylinders | Flywheel Diameter, Width, Inches | Weight, Pounds | Exhaust Opening, Inches | Piston Length, Inches | Weight, Pounds | Floor Space, Inches | Height above Floor | | Concrete Cubic Yards of |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | | Ft. | In. | |
| 50 | 33 | 1 | 62×10 | 3300 | 8 | 24 | 19,100 | 62× 81 | 9 | 11 | 7.9 |
| 60 | 40 | 1 | 66×10 | 3600 | 8 | 28 | 21,400 | 66× 87 | 10 | 1½ | 8.7 |
| 100 | 67 | 2 | 62×10 | 3300 | 8 | 24 | 25,700 | 62×109 | 9 | 11 | 11.3 |
| 120 | 80 | 2 | 66×10 | 3600 | 8 | 28 | 32,600 | 66×120 | 10 | 1½ | 12.5 |
| 150 | 100 | 3 | 62×10 | 3300 | 8 | 24 | 35,100 | 62×137 | 9 | 11 | 14.7 |
| 180 | 120 | 3 | 66×10 | 3600 | 8 | 28 | 42,400 | 66×153 | 10 | 1½ | 16.2 |
| 200 | 134 | 4 | 62×10 | 3300 | 8 | 24 | 39,500 | 62×166 | 9 | 11 | 18.1 |
| 240 | 160 | 4 | 66×10 | 3600 | 8 | 28 | 51,200 | 66×186 | 10 | 1½ | 20.0 |
| 300 | 200 | 6 | 62×10 | 3300 | 8 | 24 | 59,000 | 62×222 | 9 | 11 | 24.8 |
| 360 | 240 | 6 | 66×10 | 3600 | 8 | 28 | 64,000 | 66×252 | 10 | 1½ | 27.0 |

Note.—The shipping weights above are intended to cover shipment where engines are sent out with the average equipment.

## TABLE 181

### Fairbanks-Morse Diesel Engines

| Horse Power | Cylinders | R.p.m. | Floor Space | | | | | | Height above Floor | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | Standard and belted electric | | | Direct-connected electric engine and generator | | | | |
| | | | Ft. | | Ft. In. | Ft. | | Ft. In. | Ft. | In. |
| 40 | 1 | 300 | 8 | × | 9 7 | 8 | × | 13 7½ | 8 | 7 |
| 60 | 1 | 257 | 8 | × | 11 3 | 8 | × | 13 6 | 9 | 6 |
| 80 | 2 | 300 | 8 | × | 12 0 | 8 | × | 15 3½ | 8 | 7 |
| 120 | 2 | 257 | 8 | × | 14 7 | 8 | × | 16 8½ | 9 | 6 |
| 180 | 3 | 257 | 8 | × | 17 10 | 8 | × | 19 10 | 9 | 6 |
| 240 | 4 | 257 | 8 | × | 21 0 | 8 | × | 22 7 | 9 | 6 |
| 360 | 6 | 257 | 8 | × | 28 0 | 8 | × | 30 6¾ | 9 | 6 |

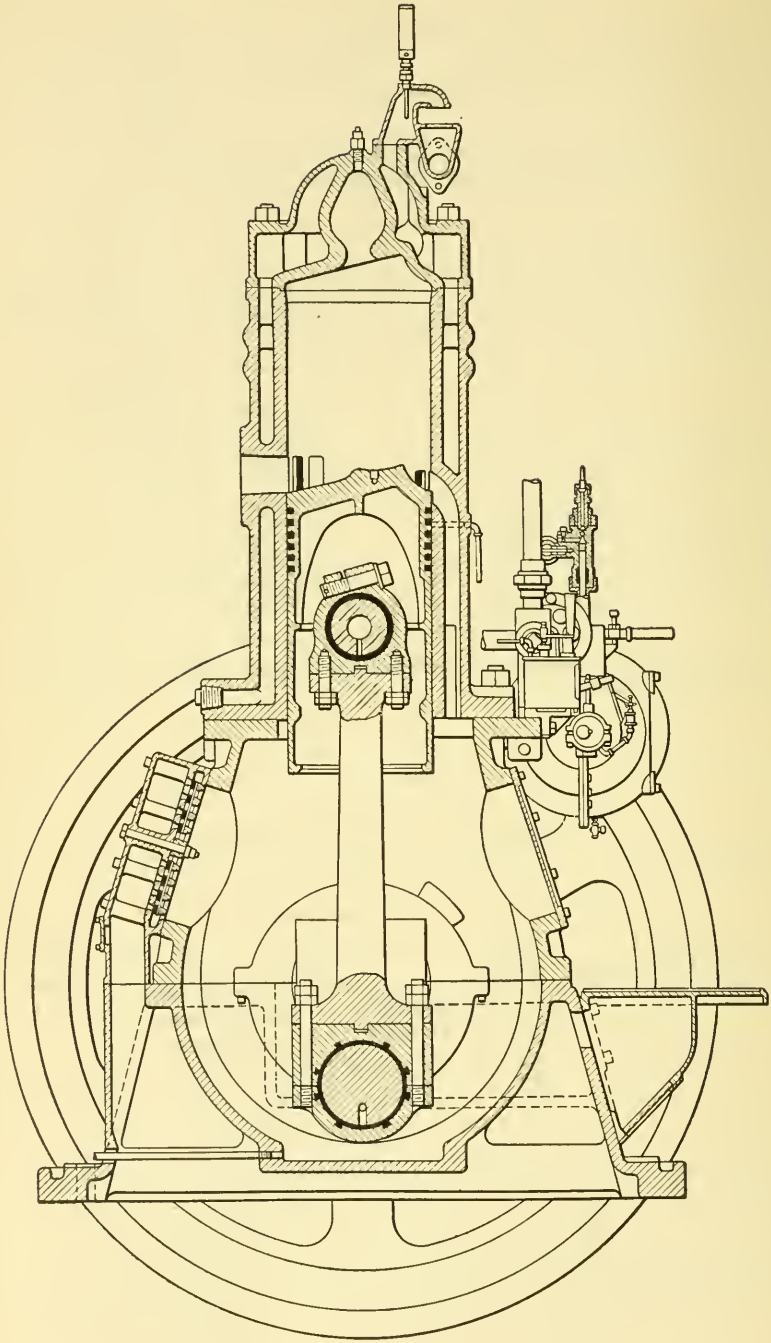

Fig. 396.—Anderson Oil Engine.

cylinder head, fuel nozzle, governor, piston head and details governing the method of ignition.

Figure 398 gives a typical fuel consumption curve of a 100-hp. oil

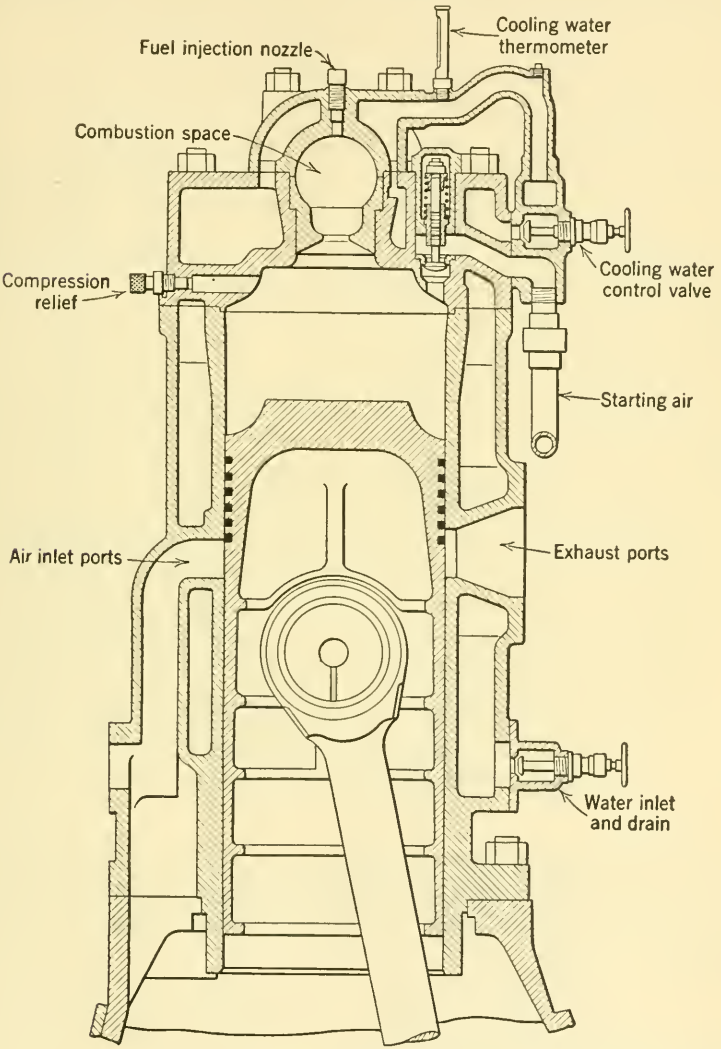


Fig. 397.—Fairbanks-Morse Oil Engine.

engine. The oil engine-driven plant, consisting of two 100-hp. oil engines, would cost with auxiliaries, including a 60-kw. generator unit, exhaust silencers, starting unit, oil storage tank, foundations, etc., about $17,500

more than the electrically driven plant consisting of two 100-hp. synchronous motors with exciters and control panels. This cost, figured on a basis of 15 per cent overhead, to include interest, depreciation and taxes would amount to $2625 per year. To make the oil engine economi-

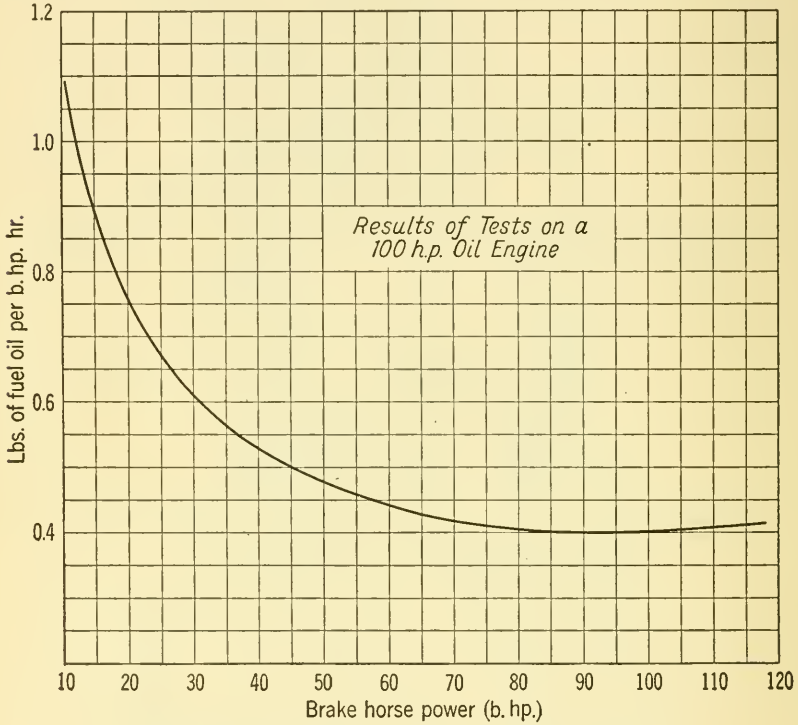


Fig. 398.—Worthington Oil Engine Economy Curve.

cal, the saving figured from a consideration of the fuel and operator costs must pay for the additional costs within a reasonable time.

Figures 396 and 397 give details of different designs of oil engine cylinders.

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